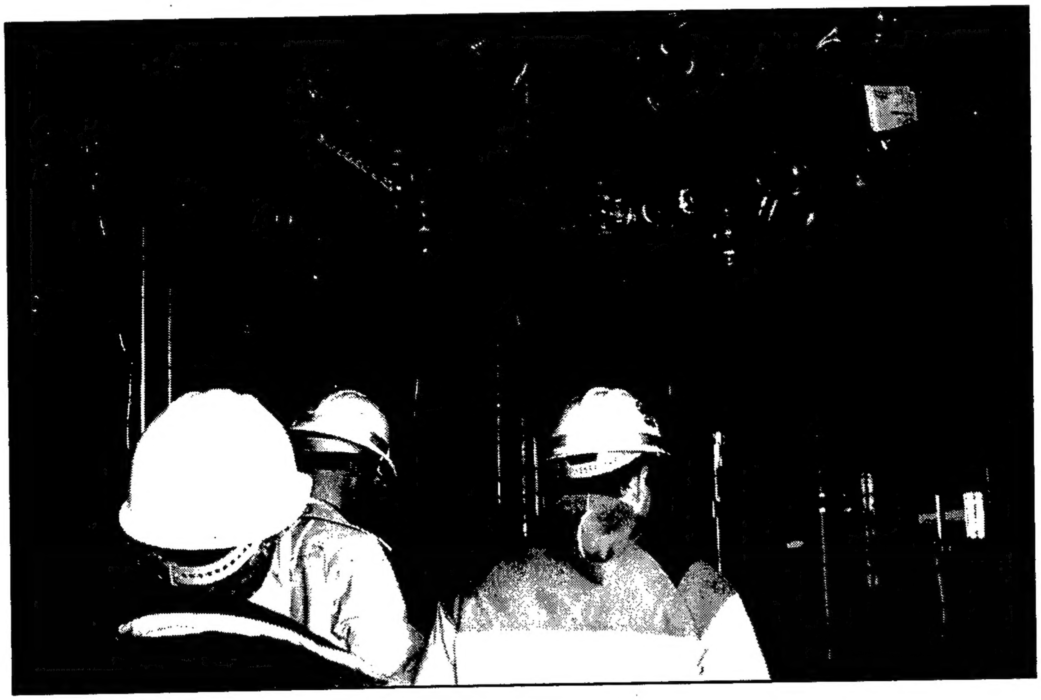
US Army Corps of Engineers
Construction Engineering Research Laboratory

USACERL Technical Report 99/20
May 1999

Acceptance Testing Procedures for Heating, Ventilating, and Air-Conditioning Systems

Dahtzen Chu, Charles L. Burton, Leland V. Speirs, Alison J. Pacheco, and Stacy Campbell



Properly operating heating, ventilating, and air-conditioning (HVAC) systems are essential for Army facilities. Operating efficiently, they conserve energy and provide a comfortable, healthy work environment. Current design and construction practices should be capable of producing functional HVAC systems, but there are no assurances of this.

Acceptance testing ensures that U.S. Army Corps of Engineers (USACE) field offices and installation Directorates of Public Works (DPWs) are receiving properly operating HVAC systems.

The objective of this work was to develop simple, accurate procedures for USACE District, Area, and Resident Office construction and engineering personnel, and DPW engineering and operations and maintenance personnel to ensure that a new facility's HVAC system is operating properly and to correct faulty existing HVAC systems. This report contains discussions on and procedures for variable air volume systems, package boilers, chillers, exhaust systems, and hydronic systems.

USER EVALUATION OF REPORT

REFERENCE: CERL Technical Report 99/20, *Acceptance Testing Procedures for Heating, Ventilating, and Air-Conditioning Systems*

Please take a few minutes to answer the questions below, tear out this sheet, and return it to CERL. As user of this report, your customer comments will provide CERL with information essential for improving future reports.

1. Does this report satisfy a need? (Comment on purpose, related project, or other area of interest for which report will be used.)

____

____

____

2. How, specifically, is the report being used? (Information source, design data or procedure, management procedure, source of ideas, etc.)

____

____

3. Has the information in this report led to any quantitative savings as far as manhours/contract dollars saved, operating costs avoided, efficiencies achieved, etc.? If so, please elaborate.

____

____

4. What is your evaluation of this report in the following areas?

a. Presentation: _______________________________

b. Completeness: _______________________________

c. Easy to Understand: _______________________________

d. Easy to Implement: _______________________________

e. Adequate Reference Material: _______________________________

f. Relates to Area of Interest: _______________________________

g. Did the report meet your expectations? _______________________________

h. Does the report raise unanswered questions? _______________________________

i. General Comments. (Indicate what you think should be changed to make this report and future reports of this type more responsive to your needs, more usable, improve readability, etc.)

____

____

____

____

____

____

5. If you would like to be contacted by the personnel who prepared this report to raise specific questions or discuss the topic, please fill in the following information.

Name: ______________________

Telephone Number: ______________________

Organization Address: ______________________

____

____

6. Please mail the completed form to:

Department of the Army
CONSTRUCTION ENGINEERING RESEARCH LABORATORY
ATTN: CEERD-IM-IT
P.O. Box 9005
Champaign, IL 61826-9005

USER EVALUATION OF REPORT

REFERENCE: CERL Technical Report 99/20, *Acceptance Testing Procedures for Heating, Ventilating, and Air-Conditioning Systems*

Please take a few minutes to answer the questions below, tear out this sheet, and return it to CERL. As user of this report, your customer comments will provide CERL with information essential for improving future reports.

1. Does this report satisfy a need? (Comment on purpose, related project, or other area of interest for which report will be used.)

___

___

___

2. How, specifically, is the report being used? (Information source, design data or procedure, management procedure, source of ideas, etc.)

___

___

3. Has the information in this report led to any quantitative savings as far as manhours/contract dollars saved, operating costs avoided, efficiencies achieved, etc.? If so, please elaborate.

___

___

4. What is your evaluation of this report in the following areas?

a. Presentation: ___

b. Completeness: ___

c. Easy to Understand: ___

d. Easy to Implement: ___

e. Adequate Reference Material: ___

f. Relates to Area of Interest: ___

g. Did the report meet your expectations? ___

h. Does the report raise unanswered questions? ___

i. General Comments. (Indicate what you think should be changed to make this report and future reports of this type more responsive to your needs, more usable, improve readability, etc.)

____

____

____

____

____

____

5. If you would like to be contacted by the personnel who prepared this report to raise specific questions or discuss the topic, please fill in the following information.

Name: ______________________________

Telephone Number: ______________________________

Organization Address: ______________________________

____

____

6. Please mail the completed form to:

Department of the Army
CONSTRUCTION ENGINEERING RESEARCH LABORATORY
ATTN: CEERD-IM-IT
P.O. Box 9005
Champaign, IL 61826-9005

REPORT DOCUMENTATION PAGE

Form Approved OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave Blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	May 1999	Final

4. TITLE AND SUBTITLE
Acceptance Testing Procedures for Heating, Ventilating, and Air-Conditioning Systems

5. FUNDING NUMBERS
4A162784
AT45
EA-XL1

6. AUTHOR(S)
Dahtzen Chu, Charles L. Burton, Leland V. Speirs, Alison J. Pacheco, and Stacy Campbell

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
U.S. Army Construction Engineering Research Laboratory (USACERL)
P.O. Box 9005
Champaign, IL 61826-9005

8. PERFORMING ORGANIZATION REPORT NUMBER
TR 99/20

9. SPONSORING / MONITORING AGENCY NAME(S) AND ADDRESS(ES)
U.S. Army Corps of Engineers (USACE)
ATTN: CEMP-CE
20 Massachusetts Avenue, NW.
Washington, DC 20314-1000

10. SPONSORING / MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES
Copies are available from the National Technical Information Service, 5285 Port Royal Road, Springfield, VA 22161.

12a. DISTRIBUTION / AVAILABILITY STATEMENT
Approved for public release; distribution is unlimited.

12b. DISTRIBUTION CODE

13. ABSTRACT (Maximum 200 words)
Properly operating heating, ventilating, and air-conditioning (HVAC) systems are essential for Army facilities. Operating efficiently, they conserve energy and provide a comfortable, healthy work environment. Current design and construction practices should be capable of producing functional HVAC systems, but there are no assurances of this.

Acceptance testing ensures that U.S. Army Corps of Engineers (USACE) field offices and installation Directorates of Public Works (DPWs) are receiving properly operating HVAC systems.

The objective of this work was to develop simple, accurate procedures for USACE District, Area, and Resident Office construction and engineering personnel, and DPW engineering and operations and maintenance personnel to ensure that a new facility's HVAC system is operating properly and to correct faulty existing HVAC systems. This report contains discussions on and procedures for variable air volume systems, package boilers, chillers, exhaust systems, and hydronic systems.

14. SUBJECT TERMS
HVAC systems
maintenance and operation

15. NUMBER OF PAGES
416

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	SAR

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std 239-18
298-102

Foreword

This study was conducted for Headquarters, U.S. Army Corps of Engineers (HQUSACE) under Project 4A162784AT45, "Energy and Energy Conservation"; Work Unit EA-XL1, "Acceptance Testing Procedure for HVAC Systems." The technical monitor was John Reiley, CEMP-CE.

The work was performed by the Energy Branch (CF-E) of the Facilities Division (CF), U.S. Army Construction Engineering Research Laboratory (USACERL). The USACERL principal investigator was Dahtzen Chu. Larry M. Windingland is Chief, CF-E; and Michael Golish is Chief, CF. The technical editor was Linda L. Wheatley, Information Technology Laboratory.

Professor Charles L. Burton, and students Leland V. Speirs, Alison J. Pacheco, and Stacy Campbell from the Department of Architectural Engineering and Construction Science, Kansas State University, contributed to this report. Appreciation is expressed to Brett Willet and Ameé Urich, also of Kansas State University, for their assistance in reviewing the acceptance testing materials.

Dr. Michael J. O'Connor is Director of USACERL.

Contents

List of Figures and Tables

Figures

Tables

1 Introduction

Background

Properly operating heating, ventilating, and air-conditioning (HVAC) systems are essential to most Army facilities. However, this does not always occur. Current design and construction practices should be capable of producing functional HVAC systems, but for a variety of reasons, there are no assurances this will happen. An inadequately functioning HVAC system, besides wasting energy and requiring increased maintenance, will be unable to provide comfortable or even healthy working conditions. A deficient working environment will almost certainly adversely affect the occupants' work performance in a facility.

Acceptance testing is the means by which U.S. Army Corps of Engineers (USACE) field offices and Army installation Directorates of Public Works (DPWs) can ensure they are receiving properly operating HVAC systems. Although major construction projects are normally the Corps' responsibility, DPWs can be responsible for small projects and rehabilitation work; these projects often have as many HVAC problems as larger ones.

Acceptance testing is not the same as testing, adjusting, and balancing (TAB). TAB occurs after an HVAC system has been installed, and is performed by a mechanical contractor or subcontractor. It involves adjusting the various components of the system, balancing air flows, testing the system's performance, and repeating these steps until the system is operating correctly. Generally, problems arise because the TAB steps were done incorrectly or not at all. Occasionally, problems are also due to incorrect design. Acceptance testing is also different from "commissioning." Commissioning, as defined by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), refers to the process of "documenting and verifying the performance of HVAC systems so that systems operate in conformity with the design intent." It includes "all members of the design, construction, and operation team; owner, designer, contractor, supplier, operator, and others as may be applicable to a given project," and "extends through all phases of a project, from concept through occupancy"

(ASHRAE 1989). Acceptance testing, however, should be considered a part of the commissioning process.

Acceptance testing should be viewed as a quality assurance step that is conducted after TAB and before commissioning, or acceptance by the customer. Its purpose is to verify the contractors' work by measuring the efficiency of the systems without adjusting them, to avoid possibly violating the warranty. Acceptance testing involves visually spot checking HVAC subcomponents, and systematically measuring, analyzing, and documenting critical energy, flow, pressure, and temperature parameters using adaptations of standard TAB trade procedures. These test results should then be compared with those in the TAB report. Major discrepancies between the two indicate potential problems. In these situations, the contractor will have to explain the reason(s) for the discrepancies. An unsatisfactory explanation will require that the contractor correct these discrepancies.

The Energy Branch, Facilities Division of the U.S. Army Construction Engineering Research Laboratory (USACERL) has an ongoing effort to develop acceptance testing procedures for the more common HVAC system components. Various preliminary stages of this effort have been documented in four USACERL technical reports (Herron, Chu, and Burton, June 1986; Chu, Burton, and Imel, December 1987; Chu, Burton, and Imel, September 1988; Chu and Imel, May 1990).

This report presents acceptance testing procedures for variable air volume systems, chillers, boilers, exhaust systems, and hydronic systems, with relevant background information on each system. Acceptance testing of HVAC systems requires a basic knowledge of HVAC principles and how HVAC systems work based on those principles, an understanding of how HVAC system components work and interact with each other, and sufficient time and resources to conduct the testing. Army personnel who conduct acceptance testing also need adequate reference materials that complement the acceptance test procedures, to properly conduct acceptance testing of HVAC systems.

Objective

The objective of this work was to give potential Army users the background knowledge to enable him or her to inspect, verify, and accept or reject an HVAC system. This report is to serve as a source of HVAC reference materials and procedures for USACE District, Area, and Resident Office engineering and construction personnel, and DPW engineering, operations, and maintenance personnel. The report emphasizes the need for acceptance testing, the short payback period of the required instrumentation, and the necessity of allocating enough well-trained manpower to conduct acceptance testing.

Approach

Discussions and meetings were held with representatives with mechanical systems expertise from HQUSACE, Division, District, Area offices, and DPWs to determine which HVAC systems most needed acceptance testing procedures. A collective decision was made to develop procedures for variable air volume (VAV) systems, package boilers, chillers, exhaust systems, and hydronic systems. A procedure for testing air supply and distribution systems had been developed earlier, and is documented in the USACERL Interim Report [IR] E-88/11 (Chu, Burton, and Imel 1988).

Many engineers and construction representatives at Corps Area and Resident Offices, and engineering and operations and maintenance personnel at DPWs do not have a technical background in or knowledge of HVAC systems. Therefore, for each of these systems, basic information about each was gathered, the necessary information to document proper installation, operation, and performance was determined, and an acceptance test procedure was developed for it. Specific sources for the information in this report appears in the bibliography section of each appendix.

Mode of Technology Transfer

It is recommended that these procedures be incorporated into USACE Proponent Sponsored Engineer Corps Training (PROSPECT) courses. Additionally, training videotapes for the systems covered in this report can be produced for use in PROSPECT courses. Currently, the acceptance test procedure for air supply and distribution systems (Chu, Burton, and Imel 1988) and its videotape are available from USACERL.

2 The Need for Acceptance Test Procedures

During CERL's research in acceptance testing of HVAC systems, one concern that arose was that field personnel at many USACE offices and installation DPWs do not have the necessary background and understanding of HVAC systems. There was also no systematic and consistent approach to verifying the quality of HVAC installation and testing. As a result, CERL developed acceptance test procedures with two criteria always under consideration: (1) they should be easy to use, even for personnel with limited knowledge of HVAC fundamentals; and (2) they should require only simple and minimal calculations. In addition, a significant amount of supporting information was also researched and provided with each procedure. The purpose for this was to provide the field personnel with a reference source they could use if they required more information on an HVAC system or its components.

Acceptance test procedures for VAV systems, package boilers, chillers, exhaust systems, and hydronic systems have been completed, and are contained in Appendices A through E. Each procedure takes the form of a "user guide" that includes an introduction to the system, a discussion of the various types of that system that may be encountered, identification of the different components that may make up that system and how they interact with each other, and procedures for acceptance testing the system. Figures and tables have been provided to illustrate the concepts and procedures being discussed. Each procedure also contains a data checklist for recording visual checks and data measurements.

Because an acceptance test procedure is a quality assurance tool used to spot check the contractors' work, the actual acceptance testing portion of each procedure is relatively short compared to the supporting documentation. The steps that are described with the checklists, however, will provide the critical amount of information necessary to determine if a system is performing acceptably. Personnel who are unfamiliar with an HVAC system should first read through all sections of its procedure to gain a better understanding of it. In this way, when

acceptance testing is performed, they will know what to look for, and where to take data measurements.

Having trained and qualified personnel does not mean that acceptance testing will always be done, however. Many Corps offices and installation DPWs do not possess the necessary instrumentation used in acceptance testing. Because the cost is not excessive ($3000 to $6000) and the payback is quick (since deficiencies are discovered and corrected earlier), Corps offices and installation DPWs are strongly encouraged to allocate funding to acquire necessary testing equipment. It can be disruptive to a facility's occupants and much more costly to repair deficient HVAC systems after they have been turned over.

Another barrier to proper acceptance testing is the fact that the Corps and installation DPW organizations, the intended users of these procedures, may often be understaffed. The lack of manpower or time to conduct testing may be overcome by the possible future implementation of data recording sensors linked to computer analysis programs. This analysis and verification can be done either in real time onsite, or later in an office. This is an idea for future HVAC analysis; it is not yet a completed project. New testing methods should be adopted as they become available.

References

American Society of Heating, Refrigerating, and Air-Conditioning Engineers, *Guideline for Commissioning of HVAC Systems* (ASHRAE 1989).

Herron, D., D. Chu, and C. Burton, *Preliminary Recommendations for Improving the Construction and Acceptance Testing of Energy-Efficient Facilities*, Interim Report [IR] E-86/05/ADA169913 (U.S. Army Construction Engineering Research Laboratory [CERL], June 1986).

Chu, D., C. Burton, and M.R. Imel, *Identification of Ways to Improve Military Construction for Energy-Efficient Facilities*, Technical Report [TR] E-88/02/ADA189632 (CERL, December 1987).

Chu, D., C.L. Burton, and M.R. Imel, *Development and Initial Evaluation of an Acceptance Testing Procedure for Air Supply and Distribution Systems in New Army Facilities*, IR E-88/11/ADA202580 (CERL, September 1988).

Chu, D., and M.R. Imel, *Field Demonstration of the Acceptance Test Procedure for Air Supply and Distribution Systems*, TR E-90/08/ADA224453 (CERL, May 1990).

Abbreviations and Acronyms

ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
DPW	Directorate of Public Works
HVAC	heating, ventilating, and air-conditioning systems
PROSPECT	Proponent Sponsored Engineer Corps Training course
TAB	testing, adjusting, and balancing
USACE	U.S. Army Corps of Engineers
CERL	U.S. Army Construction Engineering Research Laboratory
VAV	variable air volume

Appendix A: Variable Air Volume Systems

Principles, Applications, and Acceptance Testing

by
Dahtzen Chu
Energy Branch, Facilities Division
U.S. Army Construction Engineering Research Laboratory

and

Charles L. Burton
Alison J. Pacheco
Leland V. Speirs
Department of Architectural Engineering and Construction Science
Kansas State University

Contents

1 Introduction

Background

A variable air volume (VAV) system basically supplies air at a constant temperature and varies the air quantity delivered to each zone to match the change in room load. In a VAV system, an air handling unit (AHU) cools or heats air to accommodate the zone with the most extreme requirements, supplying the air through ducts to various zones. At the individual zone or space, the amount of air to be provided is regulated by dampers within a VAV box or terminal.

Use of VAV systems have been a popular energy conservation choice since the 1970s. After much experience and many customer complaints, it was learned that VAV systems needed to be continually controlled. This is in contrast to constant volume systems that are manually balanced and then left alone. Thermostatically controlled volume dampers (air valves) were used for many years, particularly in low static pressure air distribution systems. In most cases, performance was less than satisfactory. They were often a compromise to achieve lower cost than using proportioning water valves, face and bypass dampers, or other control means. However, the development of higher velocity and higher pressure systems, and nondumping diffusers and grills, combined with larger zones, higher internal heating loads, and the rising cost of energy, have made VAV systems the most popular air distribution systems specified today.

Many VAV systems cause the flow of air, and therefore, the static pressure in both the supply and return duct systems to vary as the space load changes. At any given air valve position, as the static pressure changes, airflow changes. As airflow changes, the space temperature is also changed. If the load throughout the building changed gradually and at the same time, changes in static pressure would be approximately the same at each air terminal. This seldom happens. Due to building diversity, changing solar loads, fluctuating internal loads (from people, lighting, equipment, etc.) and static pressure, airflow fluctuates in trunk ducts and duct runouts as air valve positions change. When an air terminal valve closes, the static pressure in the adjacent runout and trunk duct will increase, resulting in increased airflow through the adjacent ductwork. This

change in airflow will affect the space temperature in the new area supplied by that ductwork because a higher volume of air (usually cooled) is now flowing into the area. The space thermostat in this area will eventually sense this change in temperature, and reposition its air valve for reduced flow. This reduced flow will cause a further increase in static pressure in adjacent ductwork and increase flow even further to other air terminals. If at this time another space has an increasing load, the terminal supplying that space would open, which would reduce static pressure in adjacent ductwork, and reduce airflow to the terminals that were previously throttled back. It is apparent that space thermostats alone can never stabilize a space temperature, since almost constant temperature swings will be the result.

On many systems, airflow is proportioned by "riding the fan curve." Duct static pressure can become quite high, causing increasing airflow and compounding control problems. Therefore, a means to control and at least limit static pressure is used on many systems. Control of static pressure and airflow by using inlet or outlet dampers on fans, or variable speed drives is intended to: (1) maintain a positive pressure to prevent infiltration, (2) assure that a minimum amount of outside air is supplied, and (3) keep duct pressure within the correct operating range of the air terminals. Controlling the central fan system will not completely eliminate fluctuation of duct static pressure in adjacent runouts, but this fluctuation should now be within the "correct operating range" of the air terminals. The next section describes the three primary classes of VAV terminals.

Summary

Advantages of VAV systems:

- Low initial cost for large systems (compared to obtaining the same conditions for conventional systems) due to reduced fan sizes, ductwork, filters, and casings since the capacity is based on the peak instantaneous demand of all the spaces, instead of the sum of all the space peak demands.
- Lower operating costs due to reduced fan horsepower.
- Lower energy consumption since cooling and heating is provided only to the extent that it is required.
- Savings in mechanical space requirements due to smaller fans and ductwork.
- System is virtually self-balancing since boxes are set for maximum and minimum cfm.
- Excellent space condition controls.

Practical problems that may be encountered:

- Acoustical problems—noise generated by a terminal device varies with the static pressure across the device.
- Stratification and drafts—the air distribution patterns of conventional diffusers depend on the outlet air velocity. When airflow is reduced, the distribution pattern is changed and can cause stratification or drafts to occur.
- Unstable operation—variations in airflow cause variations in duct static pressure. As volume is reduced, duct system pressure drop is reduced and fan pressure increases. The combined effect of these two factors is higher static pressure at the terminal device as airflow is reduced. This is only a problem in pressure dependent and volume limiting systems.
- Control problems—(1) How to sense small changes in static pressure, (2) How to balance return air systems with variations in supply air, and (3) How to maintain a constant flow of outside air with variations in supply and return airflows.
- Not acceptable for some specific areas in hospitals because, at low load conditions, less air is discharged from supply outlets. This may not meet strict ventilation or humidity control requirements.
- Not especially adaptable to small volume system unless it is a low pressure system.

General applications:

- Ideal for buildings with internal spaces that have large internal heat gain.
- Most common for new institutional and office buildings where precise humidity control is not critical.
- VAV independent systems are satisfactory in schools when controls are applicable for varying loads.
- Ideal application for pressure dependent controllability is for low pressure systems with minimum load fluctuations.

Controllability Classifications for VAV Systems

Understanding the static and flow variations of VAV terminals is important to understanding how they operate. From a controllability standpoint, all VAV terminals fall into one of three classifications: (1) Pressure Dependent, (2) Volume (CFM) Limiting, and (3) Pressure Independent.

Pressure Dependent

Pressure dependent terminals do not have controls that compensate for changes in duct static pressure. Therefore, the air volume delivered depends on upstream static pressure changes. These terminals are composed of air valves or dampers in an enclosure. A change in the thermostat signal will reposition the air valve.

Because they are subject to changes in airflow, these types of terminals seldom deliver the air quantity needed to satisfy space load. They depend on the thermostat sensing the change in room temperature. The room temperature variation is brought on by airflow fluctuations. Required airflow is achieved by the repositioning of the dampers. As airflow fluctuates, the inlet static pressure also changes. These changes in static pressure cause a repositioning of the damper. If the changes in load and static pressure are great, these devices will continually oversupply or undersupply air, causing temperature fluctuations in the space as well as changes in sound levels as they "overshoot" and "undershoot."

Volume (cfm) Limiting

These terminals will compensate for change in inlet static pressure (and provide controlled airflow) only when the cfm is at a maximum. They act as a high limit to prevent the airflow from exceeding the setting of the controller. Maximum load conditions, however, exist only a few hours of the year. As a result, the terminals exhibit the same "overshooting" and "undershooting" of supply air as do pressure dependent devices, except at the maximum setpoint.

There are two types of cfm-limiting air terminal controls: (1) those using mechanical volume regulators (MVR) and (2) those using pneumatic differential pressure controllers in conjunction with either an orifice or velocity sensing probe. The MVR is a spring loaded device. It repositions to reduce airflow when static pressure exceeds the setting. The MVR is used in conjunction with an air valve upstream that is operated by a motor and thermostat.

This provides poor control. As the thermostat attempts to reduce the volume by closing the air valve, the fixed setpoint MVR will attempt to correct for the loss and maintain full volume. Only when the air valve is almost closed and has "starved" the regulator will volume begin to be reduced.

The pneumatic differential pressure controller system replaces the MVR to provide high limit control.

Below the controller's setpoint (maximum design cfm), the thermostat controls the air valve actuator. Changes in static pressure can cause the air volume to vary. The terminal is pressure dependent. If the static pressure increases to a point where it exceeds the setpoint on the controller, the controller will take control of the air valve actuator and limit the volume. In this mode, it is compensating for inlet static changes. The pneumatic pressure controllers require main control air lines to the air terminal in addition to the thermostat connection. They also consume control air at a steady rate, increasing air compressor size and operating horsepower. Cfm limiting is not often specified today.

Pressure Independent

Air terminals of this type will deliver the required amount of air to satisfy the space load regardless of changes in system static pressure. Airflow is independent of upstream static pressure changes. Overshooting and undershooting of supply air is eliminated and system stability is enhanced. Typically, this type of control will incorporate a maximum flow setting, and often a minimum flow setting, which can be factory set. The maximum flow setting is most typically used when terminal reheat is incorporated. This capability, as well as being pressure compensated, greatly reduces the amount of time and expense associated with field air balancing or the need to rebalance after building tenant changes. The mechanics of maximum and minimum settings are explained in the following paragraphs.

Space demand changes are sensed by the room thermostat whose signal resets the volume control. Therefore, these systems are frequently called Variable Constant Volume Control (VCV) or more accurately Thermostatically Resettable Constant Volume Control.

Pressure compensated air terminals are of two types: (1) those using a mechanical volume regulator and (2) those using pneumatic controllers. Air terminals using mechanical volume regulators are basically the MVR described earlier with the setpoint adjusted or reset by thermostat demand. Some have spring loaded vanes, blade dampers, or shutters in the airstream that change the effective area of the terminal as inlet static changes to maintain volume.

Several types of available pneumatic reset controls, when applied to an air terminal air valve, will provide thermostatically resettable pressure compensation. Two types are discussed here. One type uses the diaphragm principle of the differential pressure controller. Pitot tube type sensors measuring total pressure and static pressure are used to determine actual velocity pressure. Some pitot tube sensors magnify the velocity pressure 1-1/2 to 2 times. This magnification is necessary because the differential pressure controllers must have an adequate signal to operate the diaphragm assembly properly. If the air valve throttles too low, the velocity pressure signal becomes too small to be useful. Magnification of the velocity pressure signal at the sensor extends the accuracy of the differential pressure controller. Magnification of the signal is accomplished by using a venturi type sensor and the principle of "static regain."

The second type of pneumatic thermostatically reset volume control senses velocity directly using a compressed air jet principle. It does not depend on measured air velocity pressures for operation. The pneumatic output signal is linear with air velocity and is insensitive to static pressure variations. The system can accurately control at velocities lower than differential pressure controllers.

The differential pressure type controller is easier to apply, and its accuracy is usually adequate for most comfort air conditioning applications. Although both types of controllers operate at lower velocities and static pressures than resettable MVRs, they require main control air lines to the air terminal in addition to the thermostat connection. They consume control air at a steady rate; again, this increases air compressor size and operating horsepower.

Application of Controllability Classifications

Because pressure dependent terminals (PDT) are subject to change in airflow, these types of terminals seldom deliver the air quantity needed to satisfy space load. They depend on the thermostat sensing the change in room temperature due to airflow fluctuations caused by changes in inlet static to reposition the damper to provide the required airflow. If the changes in load and static pressure are great, these devices will continually oversupply or undersupply air, causing temperature fluctuations in the space. PDTs are good in some applications, but in many VAV applications, they need some compensation for change in inlet static pressure to prevent the over- and undersupply of air.

Volume limiting (VL) controls provide this compensation, but only at a maximum cfm. They act as a high setting to prevent the airflow from exceeding the

setting of the controller. A VAV system under normal conditions will only reach maximum loads a few hours out of the year. During the remainder of the year, it has the same problems as the PDT. VL is not often specified today because they require consumption of control air at a steady rate, increasing air compressor size and operating horsepower. In general, the pressure dependent system with or without VL controls is best applied where load sizes and fluctuation of load is minimal. An example would be its use in the core of an office type building that is protected from extreme solar heat transfer loads or large wind loads that can cause infiltration.

Although VL control has its advantages, it does compare with pressure independent terminals (PIT). The PIT uses a high set point and a low set point. Because of its independence from static pressure control, the PIT will deliver the required amount of air to satisfy the space load regardless of changes in system static pressure. Therefore, the problems of the PDT are eliminated. An example application would be an auditorium space in an institutional building. The space can be filled to capacity for a few hours during the day and be empty the remaining time. Note that the load fluctuations that can occur when the space goes from completely empty to completely full would be too much for a pressure dependent system to control.

Pressure compensated terminals use mechanical volume regulators (MVR) or pneumatic controllers. Resettable MVR units have a slight static pressure disadvantage due to mechanical sensing of airflow. They also will not control at as low a velocity pressure as the reset units. As mentioned earlier, many types of pneumatic controllers are available. They provide high levels of accuracy sensing pressure differentials, and are excellent for both low and high pressure systems irrespective of the size of the load fluctuations.

Control Considerations

As described earlier, VAV systems need to be continually controlled in order to meet the demands of the space. Automatic controls for VAV systems must:

- Maintain temperature differences between heating and cooling mediums at maximum economical levels.
- Conserve fan horsepower. The supply and return fans must be throttled as the system load is reduced.

- Maintain supply duct static pressure at a constant minimum level. This reduces the possibility of high pressure drops within the system, which could cause objectionable noise.*
- Maintain the volume differential between the supply and return fans and ensure that it is equal to the volume of the exhausts from the space. This is done to prevent the HVAC system from causing building pressurization problems.
- Produce system stability.

A fan is designed to discharge a given capacity of air at a given horsepower. In a VAV system, the amount of air needed depends on the needs of the space being served. Without continuous feedback control, the fan will always deliver full capacity at full horsepower and produce larger than necessary static pressures at the terminal units. Horsepower is wasted and operational savings are not realized with improper control schemes. When a control system is used to reset the fan capacity in accordance with the demands of the space, excess static pressure at the terminal units is eliminated, and the fan system horsepower adjusts to provide operational savings.

Example of a VAV System Control

Figure A-1 shows an example of a VAV system, the Modulating Zone Control System (MZCS).* This system is exemplary of a VAV system because it is designed to introduce or convert air conditioning systems to complete VAV systems using:

- Modulating damper with room thermostat that matches available airflow to zone need.
- Bypass damper that diverts and returns unneeded air.
- Leaving air temperature sensor that provides heating and cooling capacity control.
- Discriminator that can sense temperatures in up to ten locations.

---

* It must be remembered that noise is a result of pressure drops across terminal unit dampers or across diffusers. No amount of linearity of terminal unit control can reduce these effects. The room thermostat will be calling for air volumes required to meet the demand of the space, no more and no less.

* OMNIZONE, "Modulating Zone Control Systems for VAV Applications of Air Conditioning Units," Huntington Beach, CA.

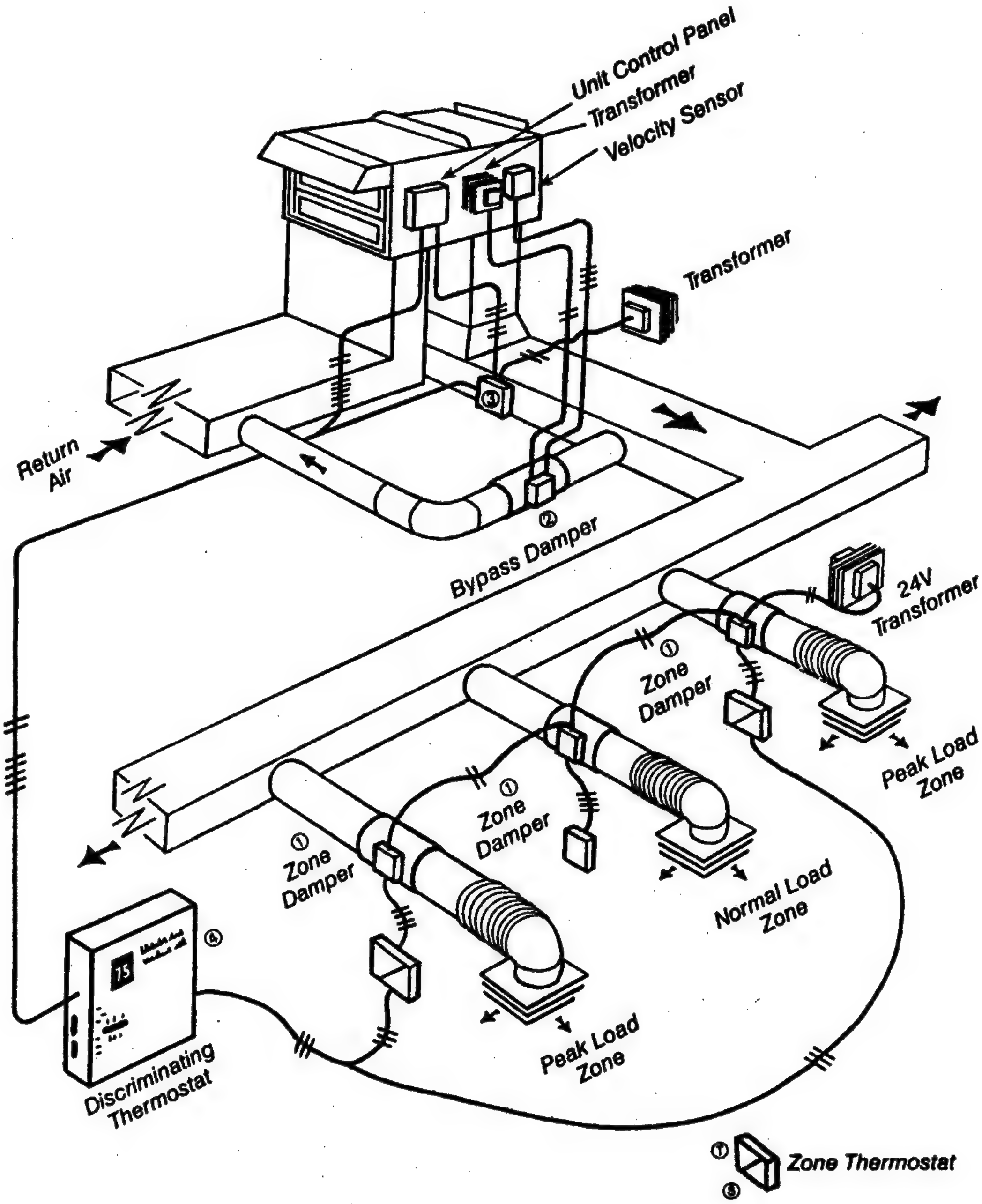


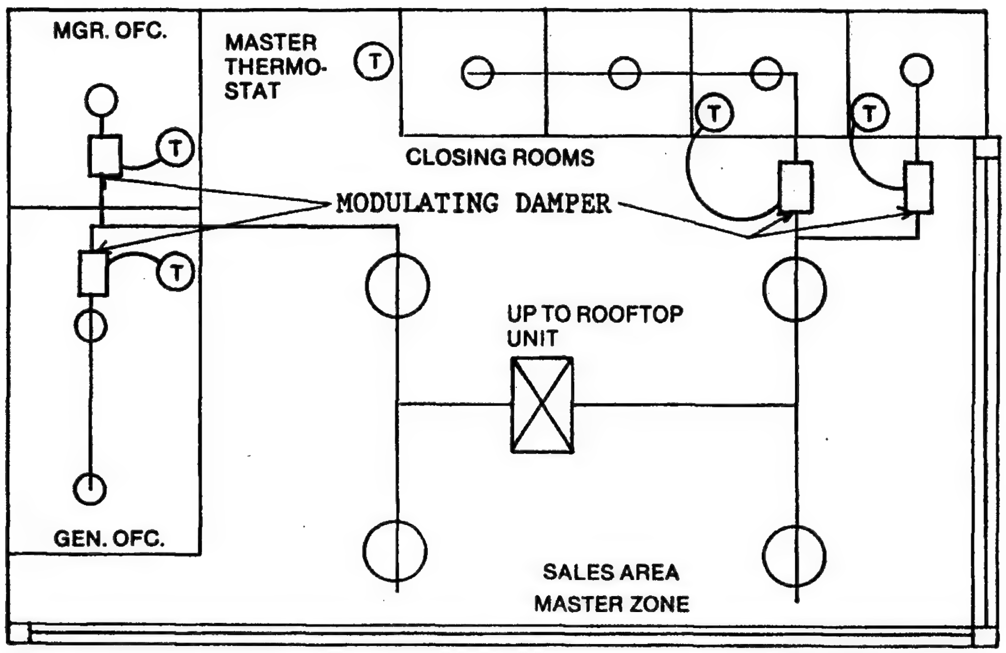
Figure A-1. Modulating Zone Control System.

Source: OMNIZONE, Huntington Beach, CA. Used with permission.

The following three examples illustrate applications of this system in different types of facilities and spaces.

Application 1

Problem: A business contains a large showroom and smaller individual offices. Individual offices might overheat and overcool.



Solution: Master-submaster (partial zoning) cooling/heating system.

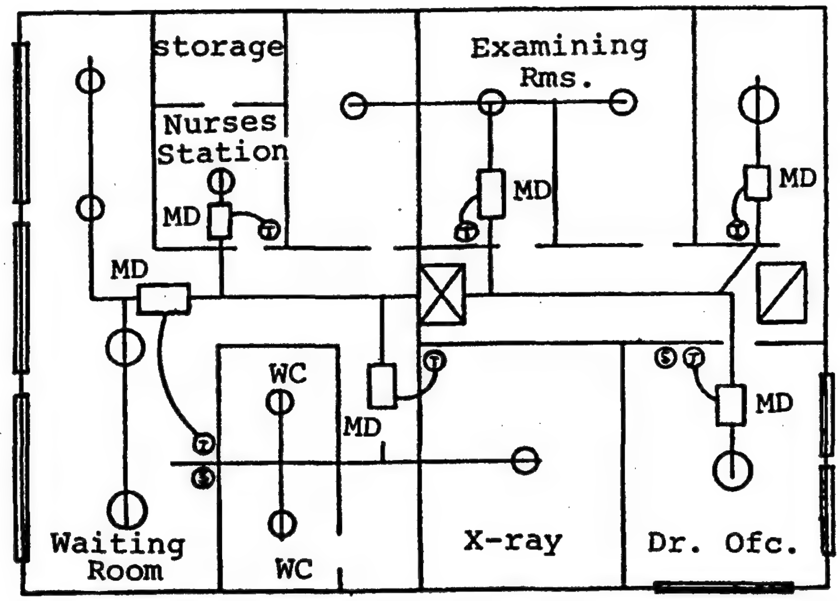
1. Set up duct system to supply air to master zone.
2. Take branch ducts to each subzone (the individual offices).
3. Install dampers on each branch duct.
4. Install thermostat in each subzone to control damper.

Install air-conditioning unit thermostat in master zone.

Operation: With the air-conditioning unit on, air will be supplied to all diffusers. As the individual offices start to be conditioned, the units will modulate to match exact room load. Excess air not required in these subzones will be shunted to the master zone. When the thermostat in the master zone is satisfied, it will shut down the refrigeration system or step out one compressor on two compressor systems. The process is similar for heating.

Application 2

Problem: Medical building with many zones, each of which vary in occupancy and load.



MD - Modulating Damper ↑N

Solution:

1. Select and install air-conditioning/heating unit to match block load.
2. Select and install bypass unit (50% of fan cfm).
3. Select convenient location for discriminator. Select sensor locations as required to monitor major loads. In this application, a sensor in the waiting room because of large west glass load and highly variable population load, and a sensor in the doctor's office because of its southeast glass load.
4. Install duct system to supply proper air quantity to each zone.
5. Install dampers in duct system to control air delivery to each zone.
6. Install thermostats in each zone.

Operation: With the discriminator set for 75 °F, the system will operate as follows. When the temperature at any zone sensor is above the setpoint, the air-conditioning unit will operate in the cooling mode. Cooling will continue to operate until all zone sensors are below 75 °F. While cooling is being supplied, each zone thermostat can control its own damper to maintain the temperature required in that zone. When all sensors are at or below 75 degrees, the discriminator will operate the system to provide ventilation. When there are no calls for cooling, and temperature at one of the sensors falls to 70 °F, the discriminator will turn on heating. Heating will operate until all sensors are at or above 70 °F, or a critical zone is again calling for cooling. During heating operation, all zone thermostats will change to the heat mode. Any zone requiring heat can be wide open. Zones with less demand will modulate to maintain the desired zone temperature. As the zone dampers modulate, fan capacity is controlled by the bypass. Compressor capacity is controlled by the leaving air temperature sensor. In this way, a variety of zone requirements can be satisfied by one air conditioning unit.

Application 3

Problem: Long building with east/west exposure. Constant volume units are a problem because one thermostat cannot furnish adequate control. As the sun's path crosses from east to west, the load of the building will follow. A system must provide adequate and cost-efficient air conditioning for the separate parts of the building as the load shifts.

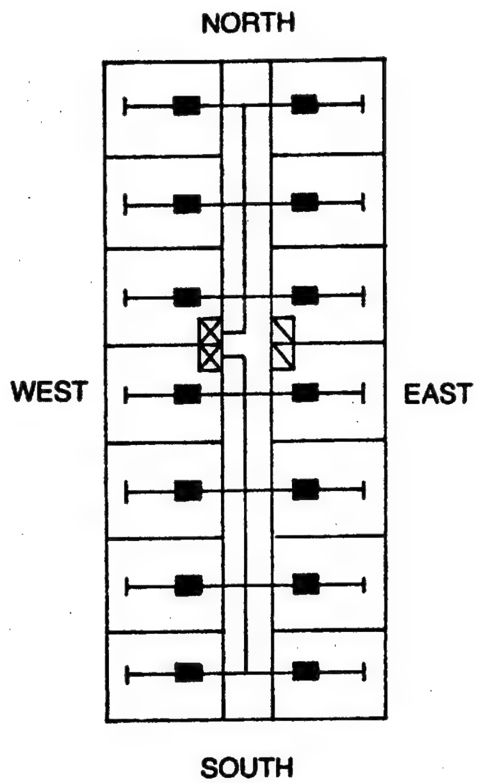
Solution: Using the MZCS will take advantage of diversity. The system will follow the sun and shift air to the side of the building with the maximum load.

1. Select air-conditioning units to match block load of building.

2. Design duct system to supply proper air quantity to each zone.

3. Select modulating dampers for each zone.

4. Select thermostat locations for each modulating damper.



5. Select bypass damper for 50 percent of unit cfm.

6. Select locations for discriminator sensors to monitor major east/west loads.

Operation: With the discriminator set for 74 °F and sensors installed on the east and west sides of the building, the discriminator will first satisfy a demand for cooling from any sensor with a temperature higher than 74 °F. The dampers will reduce air delivery to zones with lower demand for cooling. As the load shifts from east to west, the discriminator sensors will operate the package unit, and the dampers will modulate airflow to maintain required room temperatures. As the zone dampers modulate, the bypass will provide constant air volume through the unit.

Using this system of control and taking advantage of shifting sun load, the building air conditioning system can be sized around its block load rather than the sum of the peak loads.

2 Air Handling Units

VAV terminals provide conditioned air to a zone or space. Conditioned air to the VAV terminals come from air handling units typically located in a building's mechanical room. The following sections describe the basic components that are normally found in air handling units.

Basic Fan Types

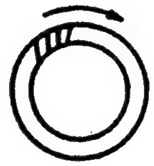
Two types of fans are generally used in HVAC work: centrifugal and axial. Centrifugal fans are most commonly used in VAV systems, particularly the forward curved and airfoil types.

Centrifugal Fans

With centrifugal fans, airflow is perpendicular to the shaft and induced by the wheel. The forward curved (FC) centrifugal fan (Figure A-2) travels at relatively low speeds and is used for producing high volumes at low static pressure. The fan will surge, but the magnitude is less than for other types. Another advantage is that it has a wide operating range. The low cost and slow speed of the FC fan are additional advantages that minimize shaft and bearing size. One disadvantage is the shape of its performance curve. It could allow overloading of the motor if system static pressure decreases. It has an inherently weak structure, and therefore is not generally capable of the high speeds necessary for developing higher static pressures.

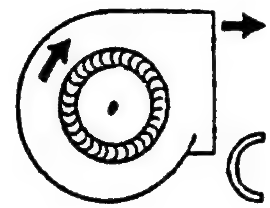
The airfoil fan (AF) (Figure A-3) is another type of centrifugal fan. It travels at about twice the speed of the FC fan. Generally, the larger the fan, the greater the efficiency. The magnitude of the AF fan's surge is also greater than that of the FC fan. Its higher speeds and bearing sizes, along with nonoverloading brake horsepower (BHP), allow higher efficiency but make proper wheel balance more critical. Also, as block-tight static pressure is approached, unstable operation may occur.

IMPELLER DESIGN



Efficiency is somewhat less than airfoil and backwardly curved bladed fans. Usually fabricated of light-weight and low cost construction. Has 24 to 64 shallow blades with both the heel and tip curved forward. Air leaves wheel at velocity greater than wheel tip speed and primary energy transferred to the air is by use of high velocity in the wheel. For given duty, wheel will be the smallest of all centrifugal types and operate at lowest speed.

HOUSING DESIGN



Scroll is similar to and often identical to other centrifugal fan designs. The fit between the wheel and inlet is not as critical as on airfoil and backwardly inclined bladed fans.

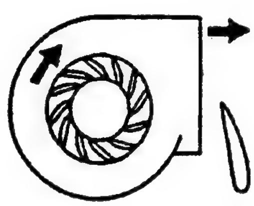
Figure A-2. Forward Curved Centrifugal Fan.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

IMPELLER DESIGN

Highest efficiency of all centrifugal fan designs. 10 to 16 blades of airfoil contour curved away from the direction of rotation. Air leaves the impeller at a velocity less than its tip speed and relatively deep blades provide for efficient expansion within the blade passages. For given duty this will be the highest speed of the centrifugal designs.

HOUSING DESIGN



Scroll type. Usually designed to permit efficient conversion of velocity pressure to static pressure thus permitting a high static efficiency. Essential that clearance and alignment between wheel and inlet bell be very close in order to reach the maximum efficiency capability.

Concentric housings can also be used as in power roof ventilators. Since there is efficient pressure conversion in the wheel.

Figure A-3. Airfoil Centrifugal Fan.

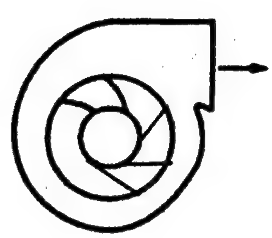
Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

Two additional types of centrifugal fans are the backwardly curved or backwardly inclined fan (Figure A-4), and the radial tip or radial blade fan (Figure A-5).

IMPELLER DESIGN

Efficiency is only slightly less than that of airfoil fans. Backwardly inclined or backwardly curved blades are single thickness. 10 to 16 blades curved or inclined away from the direction of rotation. Efficient for the same reasons given for the airfoil fan above.

HOUSING DESIGN

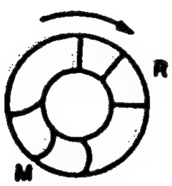


Utilizes the same housing configuration as the airfoil design.

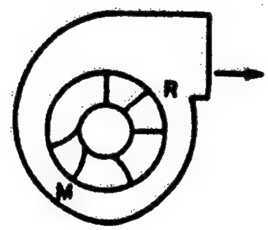
Figure A-4. Backwardly Curved or Backwardly Inclined Centrifugal Fan.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

IMPELLER DESIGN



Simplest of all centrifugal fans and least efficient has high mechanical strength and the wheel is easily repaired. For a given point of rating this fan requires medium speed. This classification includes radial blades (R) and radial tip blades (M). Usually 6 to 10 in number.



HOUSING DESIGN

Scroll type. Usually the narrowest design of all centrifugal fan designs described here. Due to less efficient wheel capabilities dimensional requirements of this housing are not as critical as for airfoil and backwardly inclined fans.

Figure A-5. Radial Tip or Radial Blade Centrifugal Fan.

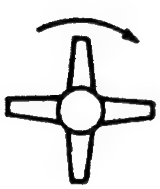
Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

Forward curved fans are stable and economical, but airfoil fans are better for discharging air into a plenum, and for multizone units and others having higher pressure through the system. The airfoil fan requires about 75 to 80 percent wide open volume, but the maximum BHP of the forward curved blade wheel is 100 percent wide open. This is a disadvantage for systems with large pressure fluctuations. Since fans operate best at peak efficiency, it is important to choose their size properly for quiet performance. For static pressures above 2 in. w.g., the backward inclined and airfoil fan are used. Below 2 in. w.g., the forward curved fan is best as far as noise is concerned.

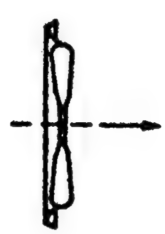
Axial Fans

In axial fans, the airflow is parallel to the shaft. Axial fans include propeller (Figure A-6), tubeaxial (Figure A-7), and vaneaxial (Figure A-8).

IMPELLER DESIGN



Efficiency is low. Impellers are usually of inexpensive construction and limited to low pressure applications. Impeller is of 2 or more blades. Usually of single thickness attached to relatively small hub. Energy transfer is primarily in form of velocity pressure.



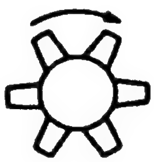
HOUSING DESIGN

Simple circular ring. Orifice plate or venturi design. Design can substantially influence performance and optimum design is reasonably close to the blade tips and forms a smooth inlet flow contour to the wheel.

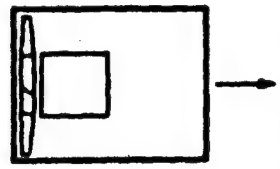
Figure A-6. Propeller Fan.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

IMPELLER DESIGN



Somewhat more efficient than propeller fan design and is capable of developing a more useful static pressure range. Number of blades usually from 4 to 8 and hub is usually less than 50% of fan tip diameter. Blades can be of airfoil or single thickness cross-section.



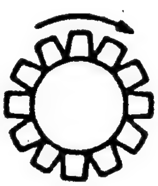
HOUSING DESIGN

Cylindrical tube formed so that the running clearance between the wheel tip and tube is close. This results in significant improvement over propeller fans.

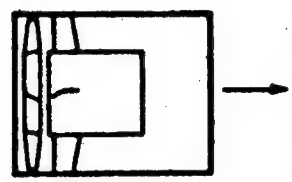
Figure A-7. Tubeaxial Fan.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

IMPELLER DESIGN



Good design of blades permits medium to high pressure capability at good efficiency. The most efficient fans of this type have airfoil blades. Blades are fixed or adjustable pitch types and hub is usually greater than 50% of fan tip diameter.



HOUSING DESIGN

Cylindrical tube closely fitted to the outer diameter of blade tips and fitted with a set of guide vanes. Upstream or downstream from the impeller. Guide vanes convert the rotary energy imparted to the air and increase pressure and efficiency of fan.

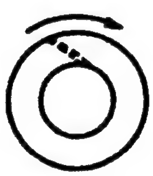
Figure A-8. Vaneaxial Fan.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

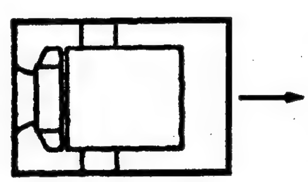
Other Fan Types

Two additional fan types that may be encountered are the tubular-centrifugal fan (Figure A-9), and the inline centrifugal duct fan (Figure A-10).

IMPELLER DESIGN



This fan usually has a wheel similar to the airfoil. Backwardly inclined or backwardly curved blade as described above. (However this fan wheel type is of lower efficiency when used in fan of this type.) Mixed flow impellers are sometimes used.

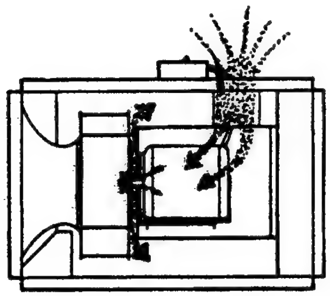


HOUSING DESIGN

Cylindrical shell similar to a vaneaxial fan housing except the outer diameter of the wheel does not run close to the housing. Air is discharged radially from the wheel and must change direction by 90 degrees to flow through the guide vane section.

Figure A-9. Tubular-Centrifugal Fan.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*



Superior aerodynamic performance is provided by deep venturi inlet combined with median-foil wheels. Casing sizes and internal baffling selected for optimum airflow. More air per dollar first-cost and more air per dollar of operating cost are provided by the high air handling efficiency. The same design details that produce maximum aerodynamic efficiency also assure quiet operation.

Figure A-10. Inline Centrifugal Duct Fan.

Source: Carnes Co., Verona, WI. Used with permission.

Fan Classes

Fans are classified according to certain construction features such as thickness of metal, type of bracing, etc. Fan classification is usually shown in the manufacturer's performance data. The TAB technician should be aware of the fan classification as this will affect whether or not the operating conditions of the fan can be altered in order to balance the system.

Fan Laws and Sizing

Shapes of performance curves for various fan types, and other information about static pressure, BHP, and rotations of the wheel per minute may be plotted on a fan curve (Figure A-11). Also, selection of fans to fit a system may be found by plotting on this curve.

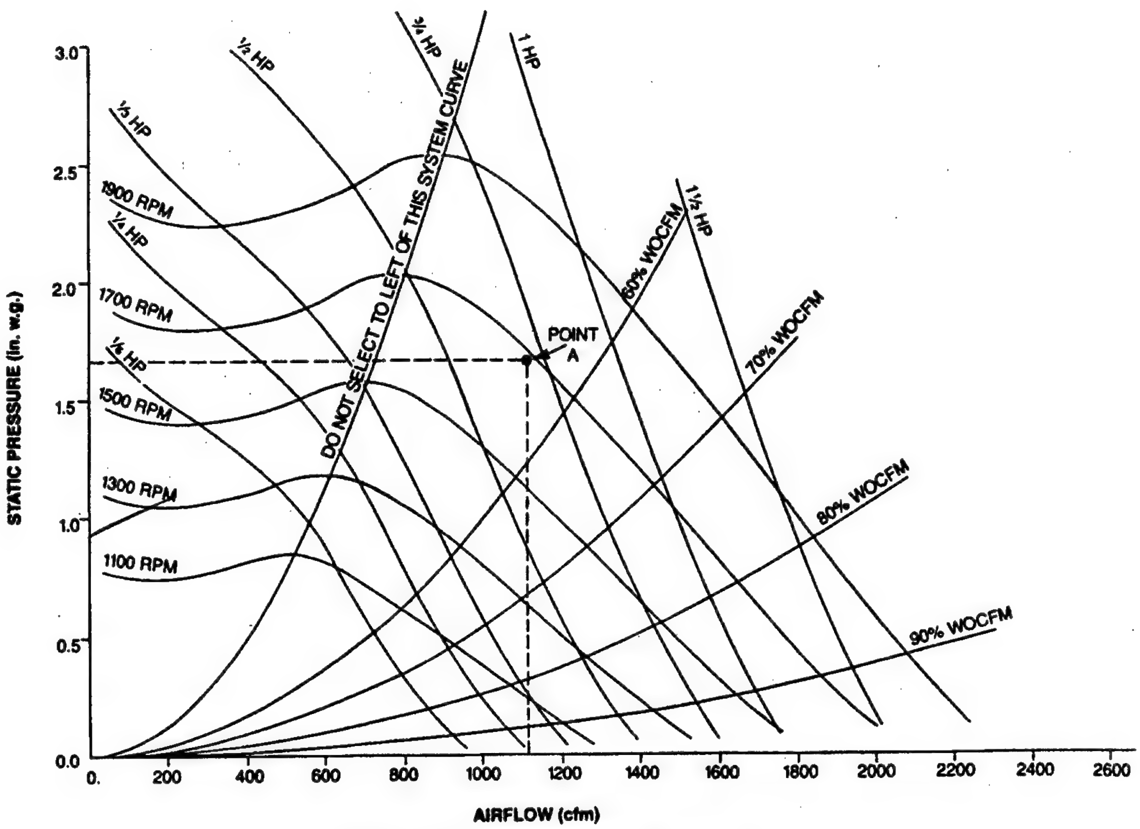


Figure A-11. Typical FC Fan Curves.

The following examples demonstrate applications of fan laws.

Fan Laws:

$$\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1} \qquad \frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3 \qquad \frac{P_2}{P_1} = \left(\frac{Q_2}{Q_1}\right)^2$$

where:

Q = airflow (cfm)
rpm = revolutions/minute
P = system pressure (in. w.g.)
bhp = brake horsepower

Example 1:

A fan must be speeded up to supply 13,000 cfm. The airflow is presently measured at 10,000 cfm at 2.0 in. w.g. static pressure. What will be the new fan speed, if the present fan speed is 660 rpm?

From $\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1}$ then $rpm_2 = 660 \times \frac{13{,}000}{10{,}000} = 858\ rpm$

Example 2:

A duct system is operating at 2.0 in. w.g. with an airflow of 10,000 cfm. If the airflow is increased to 13,000 cfm without any other change, what is the new duct system pressure?

From $P_2 = P_1 \times \left(\frac{Q_2}{Q_1}\right)^2$ then $P_2 = 2.0 \times \left(\frac{13{,}000}{10{,}000}\right)^2$

$$P_2 = 2.0 \times (1.3)^2 = 3.38\ in.\ w.g.$$

From the fan curve in Figure A-12, 3.38 in. w.g. of static pressure at 13,000 cfm requires an estimated 860 rpm. (When using this equation, the system pressure can be in terms of either total pressure or static pressure.)

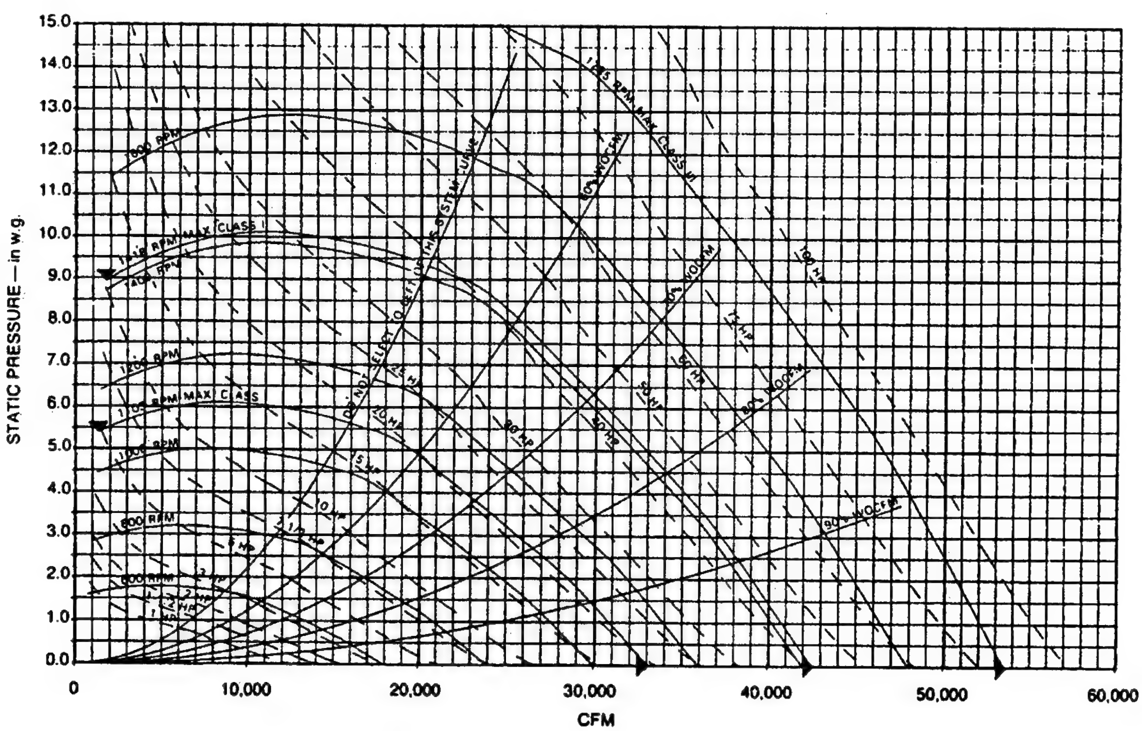


Figure A-12. Fan Curves.

Example 3:

The same system used in Example 2 has a 5 HP motor operating at 4.0 bhp. Find the bhp that would be required if the airflow was increased to 13,000 cfm.

From $bhp_2 = bhp_1 \times \left(\frac{Q_2}{Q_1}\right)^3$ then $bhp_2 = 4.0 \times \left(\frac{13{,}000}{10{,}000}\right)^3$

$$bhp_2 = 4.0 \times (1.3)^3 = 8.79\ bhp$$

The 5 HP motor would be inadequate, and a 10 HP motor would be required.

The next three examples show how fans are sized. Before proceeding further, some terms used in the examples will be defined first:

SP	Static Pressure:	The normal force per unit area that would be exerted by the moving air on a balloon immersed in it if it were carried along by the air.
ISP	Internal Static Pressure:	The sum of the pressure drops across components inside air handling units such as filters, cooling and heating coils, hot and cool deck dampers, etc.
ESP	External Static Pressure:	The sum of the pressure drop across components external to the air handling unit such as terminal boxes, elbows, diffusers, volume dampers, and all other friction causing elements in the duct system.

SPs Static Pressure (suction)

SPd Static Pressure (discharge)

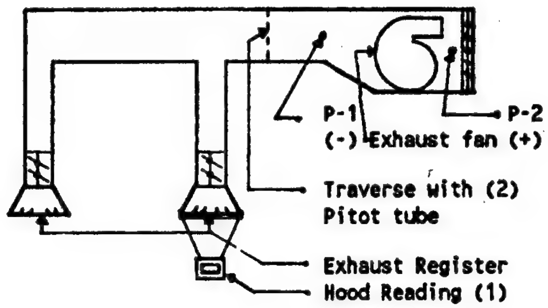
ISPcc Internal Static Pressure (cooling coil)

ISPhc Internal Static Pressure (heating coil)

ISPf Internal Static Pressure (filters)

Fan SP = SPd - SPs

Example 4: Single Zone Exhaust System



Find:

1. P-1 Neg. (Pressure required to move air from face of exhaust grill to P-1.)
2. P-2 Pos. (Pressure required to move air through discharge louver. P-2 will be 0 in. w.g. depending on discharge configuration.)
3. cfm (Determined by hood reading at exhaust grill and pitot tube traverse.)
4. Static Pressure (Taken with pitot tube.)
5. Fan Horsepower

P-1 = SPs = -.64 in. w.g.

P-2 = SPd = .44 in. w.g.

Fan SP = SPd - SPs = .44 - (-.64) = 1.08 in. w.g.

From hood and pitot tube readings, cfm was found to be 6,509

By interpolation on Table A-1, fan bhp = 2.47.

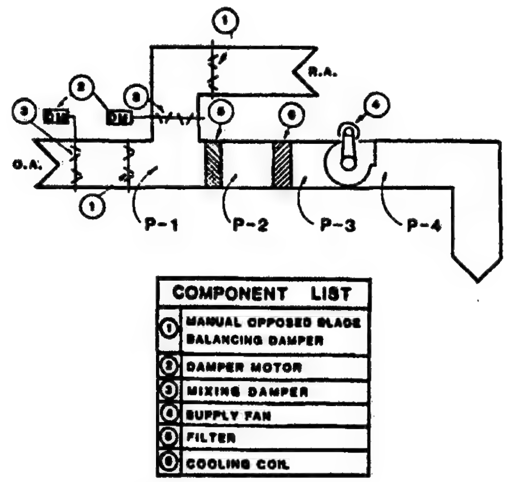
Table A-1. Fan Ratings.

VOL	OUT VEL	VEL PRES	0.125	S P	0.250	S P	0.375	S P	0.500	S P	0.625	S P	0.750	S P	0.875	S P	1.000	S P	1.250	S P	1.500	S P	1.750	S P	2.000	S P
CFM	FPM	IN H_2O	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
2264	800	0.04	398	0.10	456	0.15	507	0.21	557	0.26	606	0.32	656	0.40	703	0.47	747	0.55								
2547	900	0.05	434	0.13	487	0.19	536	0.25	578	0.30	624	0.37	669	0.44	712	0.51	755	0.60	835	0.78						
2830	1000	0.06	472	0.17	519	0.23	565	0.29	606	0.36	645	0.42	686	0.49	727	0.57	767	0.65	843	0.83	916	1.03				
3113	1100	0.08	510	0.21	552	0.27	595	0.34	636	0.42	675	0.49	708	0.56	746	0.63	782	0.71	855	0.89	924	1.10	981	1.31		
3396	1200	0.09	549	0.26	587	0.33	627	0.40	666	0.48	702	0.56	738	0.64	768	0.71	802	0.79	870	0.97	936	1.17	999	1.39	1062	1.63
3679	1300	0.11	588	0.32	624	0.39	661	0.47	697	0.55	731	0.64	765	0.73	798	0.81	825	0.89	888	1.07	950	1.28	1012	1.48	1070	1.71
3962	1400	0.12	629	0.39	662	0.46	696	0.54	729	0.63	762	0.72	794	0.81	828	0.91	856	1.01	909	1.17	967	1.37	1026	1.58	1083	1.82
4245	1500	0.14	668	0.46	700	0.54	730	0.62	762	0.72	794	0.81	825	0.91	854	1.01	884	1.12	936	1.30	989	1.50	1043	1.71	1097	1.94
4528	1600	0.16	709	0.55	739	0.63	767	0.72	796	0.81	827	0.91	856	1.02	884	1.13	912	1.23	967	1.46	1013	1.64	1063	1.88	1114	2.08
4811	1700	0.18	749	0.65	778	0.74	806	0.83	832	0.92	860	1.03	888	1.14	916	1.25	942	1.36	994	1.59	1044	1.82	1087	2.01	1134	2.25
5094	1800	0.20	790	0.75	818	0.85	843	0.95	868	1.06	894	1.15	921	1.26	948	1.38	973	1.50	1023	1.74	1073	1.99	1115	2.21	1157	2.43
5377	1900	0.23	830	0.88	857	0.98	882	1.08	908	1.19	930	1.29	955	1.40	980	1.53	1005	1.65	1053	1.90	1100	2.16	1146	2.42	1185	2.64
5660	2000	0.25	872	1.01	897	1.12	921	1.23	944	1.33	966	1.44	989	1.56	1014	1.68	1038	1.81	1084	2.08	1129	2.34	1173	2.61	1217	2.89
5943	2100	0.27	913	1.16	937	1.27	960	1.39	982	1.50	1004	1.61	1025	1.73	1048	1.85	1071	1.99	1116	2.28	1160	2.54	1202	2.82	1245	3.12
6226	2200	0.30	954	1.32	977	1.44	999	1.56	1021	1.68	1042	1.80	1062	1.91	1083	2.04	1104	2.17	1148	2.46	1191	2.75	1231	3.04	1272	3.34
6509	2300	0.33	995	1.50	1017	1.62	1039	1.75	1058	1.87	1080	1.99	1100	2.12	1119	2.24	1139	2.38	1181	2.67	1222	2.97	1262	3.28	1301	3.58
6792	2400	0.36	1037	1.70	1067	1.82	1079	1.95	1099	2.08	1118	2.21	1137	2.34	1156	2.47	1176	2.60	1215	2.90	1255	3.20	1293	3.52	1331	3.84
7358	2600	0.42	1120	2.13	1139	2.26	1159	2.40	1178	2.55	1196	2.68	1214	2.82	1231	2.97	1248	3.10	1284	3.40	1321	3.72	1358	4.06	1393	4.40
7924	2800	0.49	1204	2.64	1221	2.78	1239	2.93	1257	3.08	1274	3.23	1291	3.38	1308	3.53	1324	3.69	1356	3.99	1389	4.32	1424	4.67	1458	5.03
8490	3000	0.56	1287	3.23	1303	3.38	1320	3.53	1337	3.70	1353	3.86	1370	4.02	1385	4.18	1401	4.34	1431	4.67	1461	5.00	1492	5.36	1526	5.73
9056	3200	0.64	1371	3.90	1386	4.05	1401	4.21	1417	4.39	1433	4.56	1448	4.74	1464	4.91	1478	5.08	1507	5.43	1536	5.77	1563	6.13	1593	6.51
9622	3400	0.72	1455	4.66	1469	4.82	1483	4.99	1498	5.16	1513	5.35	1528	5.54	1542	5.72	1556	5.91	1583	6.27	1611	6.64	1637	7.00	1664	7.39
10188	3600	0.81	1539	5.51	1552	5.68	1566	5.85	1579	6.04	1594	6.24	1608	6.43	1621	6.63	1635	6.82	1661	7.20	1687	7.59	1713	7.99	1737	8.37
10754	3800	0.90	1623	6.46	1636	6.64	1648	6.82	1661	7.01	1674	7.21	1688	7.42	1701	7.63	1714	7.84	1740	8.26	1764	8.65	1788	9.06	1813	9.48
11320	4000	1.00	1707	7.52	1719	7.70	1731	7.89	1743	8.09	1755	8.29	1769	8.52	1781	8.74	1794	8.96	1818	9.39	1841	9.80	1865	10.24	1888	10.68

Pressure class limits:

Class	Maximum RPM
I	1550
II	2140

Example 5: Single Zone System



Find:

P-1 = ESPs = .53 in. w.g.

ISPf and ISPcc: pressure drops available from manufacturer's data

P-2 = ISPf + ESPs = .08 + .53 = .61 in. w.g.

P-3 = ISPcc + ISPf + ESPs = (-.18) + (-.08) + (-.61) = -.87 in. w.g.

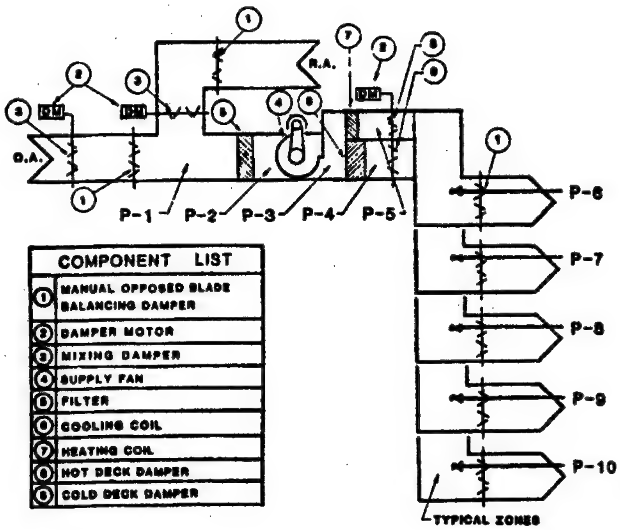
P-4 = ESPd = 1.15 in. w.g. (measured with pitot tube)

cfm = 7,924 (determined by hood and pitot readings)

Fan SP = (P-4) - (-P-3) = 1.15 - (-.85) = 2.00 in. w.g.

From Table A-1, fan bhp = 5.03.

Example 6: Multizone System



cfm = 10,188, amount of air required to supply zones 6-10

P-1 = -.40

P-2 = ISPf + (P-1) = (-.08) + (-.40) = -.48 in. w.g.

P-3 = 1.15 in. w.g.

P-4 = 1.15 - ISPcc = 1.15 - .15 = 1.00 in. w.g.

P-5 = 1.15 - ISPhc = 1.15 - .05 = 1.10 in. w.g.

In cooling mode (readings for P-6 through P-10 are obtained by pressure gauge measurements):

ESPsys6 = (P-4) - (P-6) = 1.0 - .32 = .68 in. w.g.

ESPsys7 = (P-4) - (P-7) = 1.0 - .25 = .75 in. w.g.

ESPsys8 = (P-4) - (P-8) = 1.0 - .30 = .70 in. w.g.

ESPsys9 = (P-4) - (P-9) = 1.0 - .10 = .90 in. w.g.

ESPsys10 = (P-4) - (P-10) = 1.0 - .20 = .80 in. w.g.

FAN SP = (P-3) - (P-2) = 1.15 - (-.48) = 1.63 in. w.g.

By interpolation on the fan rating table, fan bhp = 7.79.

The calculations show that P-9 has developed the largest static pressure required in the system. Therefore, fan size is based on the static pressure required at P-9, and use of the cooling coil (wet). All remaining systems require balancing with manual dampers.

Fan Curves Vs. System Curves

System resistance curves or system curves are a plot of cfm vs. static pressure in a system. This shows a graphical representation of the system's resistance to air flow. Each system will have its own system curve that is represented by a single line. This curve will remain unchanged until there is a change to the system, such as dirt or moisture buildup, or a change in position of the outlet dampers.

When the system curve and a fan performance curve are plotted together, the intersection of the two curves will be the operating point of that system. The figure below shows a typical system curve plotted with two fan curves. This example illustrates the effects of a 10 percent increase in fan speed without a change to the system itself. The operating point moves upward along the system curve resulting in an increase in both cfm and static pressure.

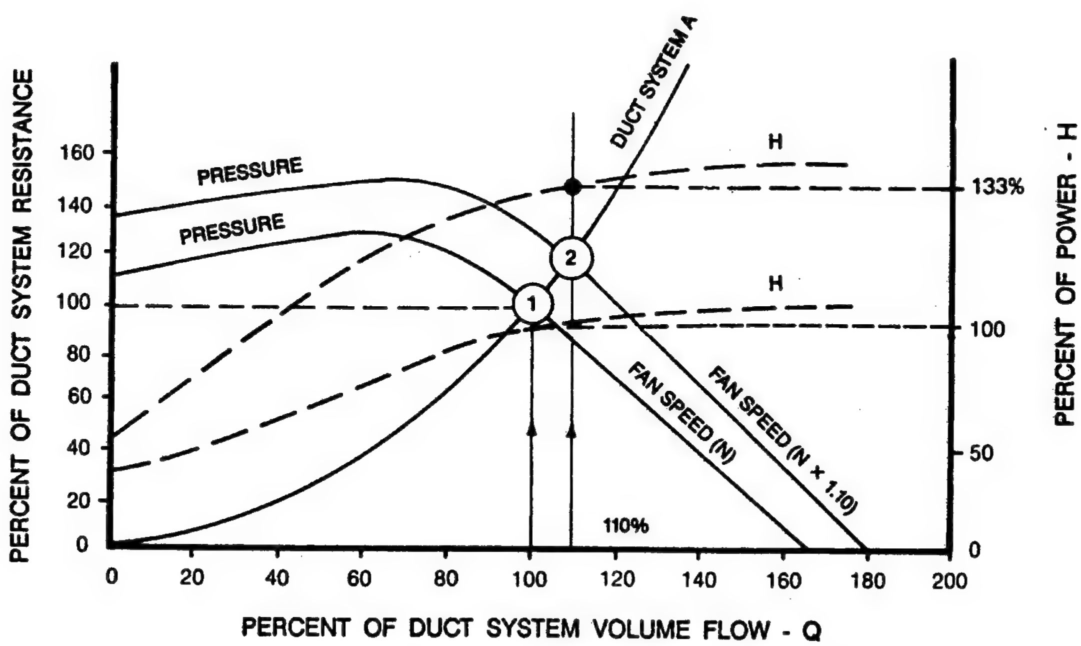


Figure A-13. Typical System Curve With Two Fan Curves.

Environmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

However, if a change is made to the system that will shift the whole system curve to the right, the operating point will move downward along the fan curve. This results in an increase in cfm with less static pressure in the system.

The fan and system curves can be used to help troubleshoot different problems that may occur in the system. They can also be used to model the effects of different changes to the system or fan. This helps in predicting what changes will produce the best results. This method, however, does not produce exact results. Therefore, when searching for exact answers, the appropriate fan laws must be used.

Fan Discharge Control

The four common methods of controlling the effects of any fan: (1) discharge damper control, (2) inlet vane control, (3) variable pitch control, and (4) speed control. Figure A-14 shows the approximate power savings that can be obtained by reducing air quantities for the four methods of capacity control.

From a power consumption standpoint, variable speed motors and blade pitch control are the most efficient. Inlet vanes save some power, while discharge dampers throttling at the fan save little. From a first-cost standpoint, dampers are the least costly. Inlet vanes and blade pitch control follow, with variable speed motors being the most expensive.

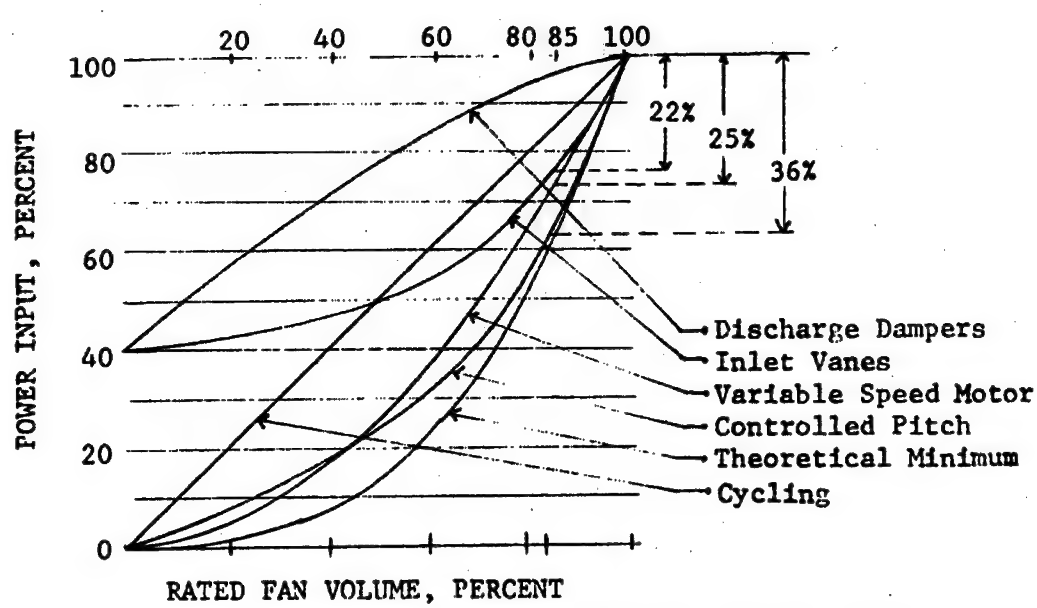


Figure A-14. Power Savings for Four Methods of Capacity Control.

Discharge Air Dampers

Discharge air dampers are installed to add resistance at the fan. The purpose of the discharge air damper arrangement is to create an excess pressure drop near the fan, thus permitting smaller pressure drops at the terminal units. The amount of air delivered to the terminal units depends on the pressure built up at the fan. When air delivery has to be reduced at the terminal unit, the terminal unit air dampers will throttle it down. It is impractical to have the terminal units serve as the sole means of throttling air supply. For example, terminal units should not have to throttle down 3 in. w.g. when all they were designed to throttle is 1 in. w.g. So, the pressure over and above what is actually needed at the terminal units is throttled down by the discharge air dampers before it even enters the rest of the system. Because of the initial throttling, there is less noise at the terminal units. With an initial pressure drop at the fan, there is more ductwork to aid in sound attenuation prior to discharge.

Sizing of discharge air dampers should be done with great care. There are many rules-of-thumb, but the recommended procedure is to size the discharge air dampers for a wide open pressure drop of from 7 to 10 percent of the system pressure.

Because discharge air dampers waste horsepower, they should not be used to control VAV systems if operational efficiency is desired. Figure A-11 shows that their efficiency is not adequate to warrant their use for economic operation purposes.

Variable Inlet Vanes

The most commonly used method of controlling fan capacity on VAV systems is variable inlet vanes. Inlet vanes, often referred to as pre-rotation vanes, cause the air to swirl before it encounters the fan wheel. The fan wheel cannot "grip" the air as well and consequently, capacity is reduced more efficiently than with discharge damper control. Excess pressure is not created and wasted. Figure A-15 shows examples of an inlet vane type system. The fan inlet vanes are positioned by an actuator in response to a signal received from the system static pressure receiver-controller.

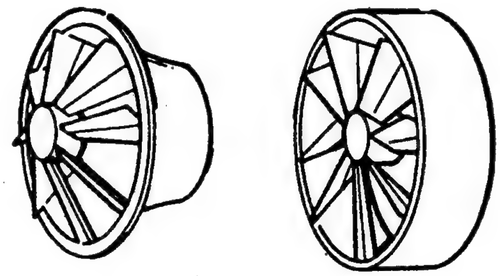


Figure A-15. Examples of Inlet Vane System.
Modification from AMCA, Publication 201-90. Used with permission.

The static pressure transmitter, through the receiver-controller and actuator, repositions the inlet vanes to maintain a relatively constant duct pressure at the point of sensing. As the terminal unit dampers throttle, the characteristic curve (resistance curve) shifts. The static pressure transmitter senses this shift, and throttles the inlet vanes accordingly. The fan curve is shifted, and a new operating point is established. The new operating point will depend on where the terminal units that are being throttled are located in the system. Fan discharge pressure will not remain constant since the location of the pressure transmitter is at the end of the system.

Care must be exercised in selecting the fan. It is important for the fan to be able to be throttled to the near minimum flow required without becoming unstable. Systems that can be throttled to near shut-off must often be equipped with a fan bypass to permit a minimum flow through the fan at all times.

Variable Pitch Blades

Variable pitch axial-flow fans deliver an amount of air in accordance with the pitch of the fan blades. As more or less air is needed in the system, an actuator positions the pitch of the fan blades accordingly. The positioning of the blades is similar to the positioning of the inlet vanes. However, the fan is always spinning while inlet vanes remain stationary. The degree to which the blades are pitched determines how much air can be "gripped" and passed on into the system.

Variable Speed Drives

Various ways to control fan speed include variable speed motors, magnetic couplers, and fluid drive systems. Fluid drive units use hydraulic fluid for

transmitting power. Magnetic coupling models use interacting magnetic fields to transmit power. Another method is to use exhaust steam, when readily available, to drive the fan with a steam turbine. The most common method, however, is motor speed control.

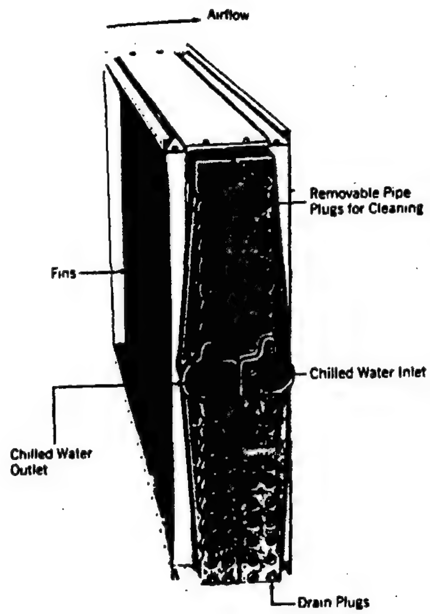
Adjustable speed drives (ASDs) are devices that vary the speed of a motor to match the load being put on the motor. Many types of ASDs are available, including mechanical (eddy current drives, variable-ratio pulley, and hydraulic drives), direct current (DC motors), and electronic. Although mechanical drives and DC motors have been applied extensively in industrial settings, they are seldom used in commercial buildings for economic or technical reasons. The mechanical variable-ratio pulley is applicable to commercial buildings (from 5 to 125 horsepower), but space requirements and mechanical problems usually make commercial applications impractical. DC motors comprise a mature technology, but they are expensive and have a reputation for high maintenance costs. The electronic load-commutated inverter has also been used in industry, but it is not an energy-conscious choice for commercial buildings.

Frequency operated adjustable speed drives are most commonly used for variable fan speed control today. Fan motor speed control is accomplished by mechanically, electrically, or hydraulically varying fan rpm in response to the signal from the pressure transmitter in the system. The transmitter/receiver-controller arrangement varies fan speed to maintain a constant duct pressure at the transmitter.

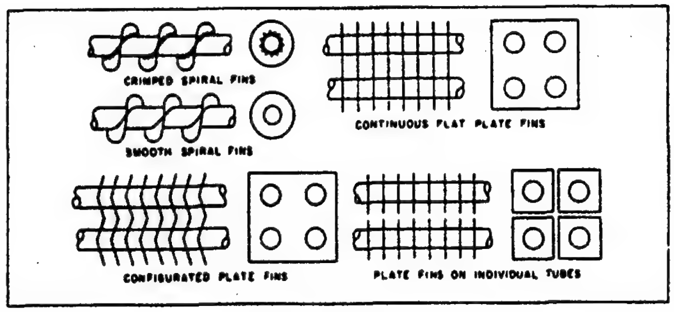
Heating and Cooling Coils

Heating and cooling coils are simply heat exchangers between a heating or cooling medium and the air stream. Heating mediums available for heating coils are steam, hot water, or electricity. Steam and hot water coils consist of banks of copper tubing surrounded by sheets of corrugated fins that guide the air toward the tubing to maximize the heat transfer surface in contact with the air. Figure A-16 is a four-row cooling coil with double-tube serpentine circuiting.

A boiler is required to produce the steam or hot water for these types of coils, which in turn requires piping from the boiler to the AHU. Thus, the steam or hot water coils are economical only for medium and large-size installations, and become a more and more attractive option as the number of AHUs served increases.



Four-row cooling coil with double-tube serpentine circuiting. *(Courtesy The Trane Company. LaCrosse. Wis.)*



TYPES OF FIN-COIL ARRANGEMENTS

Figure A-16. Four-row Cooling Coil With Double-tube Serpentine Circuiting.

A preheat coil is used to raise the outside air temperature to 55 °F before it gets to the AHU when the outside air is below 32 °F. Another special heating coil, known as a reheat coil, is sometimes placed downstream from the cooling coil for applications where humidity control is critical, such as in hospitals, laboratories, and some industries. The cooling coil dehumidifies the air to a precise point, and then the reheat coil warms it back up to the necessary temperature.

Cooling coils may carry either chilled water or refrigerant gas. The arrangement could consist of a single coil section or a number of individual coil sections built up into banks. The coil assembly will usually include an air cleaning means to protect the coil from accumulation of dirt, and to keep dust and foreign matter out of the conditioned space. Cooling coils for water or for volatile refrigerants most frequently have aluminum fins and copper tubes, although copper fins on copper tubes, and more rarely, aluminum fins on aluminum tubes are also used.* The diameter of the tubes can vary from ¼ to 1 in. The fin spacing should be chosen for the duty to be performed, with special attention being paid to air

* Approximately 90 percent of common HVAC coils are copper tube with aluminum fins due to cost, weight, and environment. There is always a difference in heat transfer between metals, but it is an insignificant amount.

friction, possibility of lint accumulation, and (especially at lower temperatures) the consideration of frost accumulation. The fins are generally spaced 3 per inch up to 14 per inch.

Coil capacity can be controlled without using a control valve. In Figure A-17, the face and bypass damper is actually two dampers linked together. When full heating (or cooling) is required, the damper section in front of the coil face is full open and the damper section in the bypass is shut. All the air passes through the coil. As the room demand for the coil capacity diminishes, a room thermostat signals a motor to move the face dampers toward the closed position while moving the bypass dampers to a more open position.

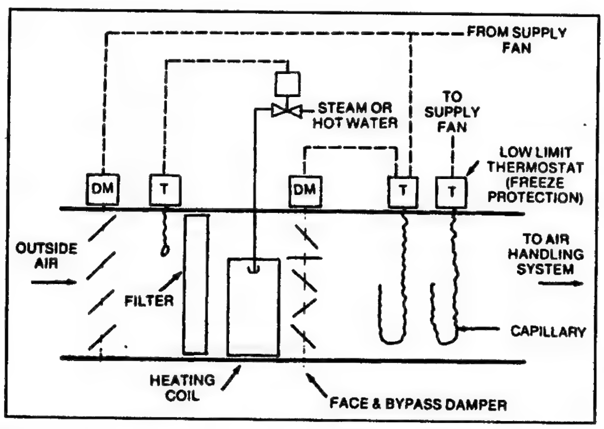


Figure A-17. Coil Capacity Controlled by Two Dampers Linked Together.

Reprinted with permission from the 1995 *ASHRAE Applications Handbook.*

Direct Expansion Coil Circuiting for Variable Air Volume Systems (DX/VAV)

Direct expansion (DX) cooling coils are thermodynamically complex. Sensible and latent heat exchange occur on the inside and outside surfaces of the coils. Mixed or all outside air flowing across the coils is both sensibly and latently cooled; this causes moisture to condense on the coil surface. Inside the refrigerant tube (cooling coil), sensible and latent heating occur as refrigerant is evaporated, and superheated. Superheated vapor (refrigerant vapor) is characterized by the actual pressure of the vapor being lower than the saturation pressure at a given temperature, and the actual temperature of the vapor being higher than the saturation temperature of the vapor. Superheating occurs beyond the saturated vapor phase, and it is very important to keep the refrigerant in this superheated phase until it gets to the compressor. If a mix between saturated vapor and saturated liquid exists in the line prior to entering the compressor, liquid refrigerant will dilute compressor oil, robbing the compressor of vital lubrication.

If left in the compressor, liquid refrigerant can cause oil foaming when the compressor restarts, and foaming hampers delivery of oil to critical crankshaft and journal bearings.

Sensible and latent heat do not occur in a linear fashion along the outside of the coil surface, so the coil may not be uniformly wetted. On the inside surface, as liquid and vapor refrigerant (mix) are forced through the tube, a pressure drop results and lowers the refrigerant boiling point.

Chilled water gets warmer as it goes down the tubes, but the refrigerant actually cools in this process. Only after all the refrigerant is completely evaporated can superheating begin to warm the vapor. The point where superheating begins also affects the coil capacity and performance because it affects the pressure drop which is not always uniform.

Even though excellent refrigerant piping practices are followed in most installations of DX/VAV split systems, some systems become very unstable, especially at part load conditions. Some of the common problems that have occurred include erratic thermal expansion valves, continued compressor cycling, coil frosting, poor temperature control, and the return of liquid refrigerant to compressors. In some severe cases, compressors can be destroyed. Considering all of these problems, suppose a system was designed with identical equipment, employing similar controls and prudent piping and installation practices, and problems still occur, what separates good and poor DX/VAV systems? The difference could be the internal circuiting of the DX cooling coil.

A distributor is the device that uniformly transfers or distributes refrigerant from the thermal expansion valve to each circuit. There is only one distributor for each expansion valve. Since the expansion valve bulb senses the degree of superheat for all the circuits on that distributor, it is unaware of any differences between circuits. Therefore, the refrigerant must be distributed uniformly to all circuits.

When the distributor's maximum MBh/circuit is insufficient to meet design load, or minimum compressor loading is less than the distributor's minimum MBh/

circuit, the coil must be divided into separate sections or "splits."* Each section is fed by one distributor.

The coil can be split in three ways to satisfy the needs of a specific design. This is where the engineer must use a knowledge of the environment in which the system will operate and select accordingly. Following is a brief description of each kind of split system and where they are best used. The three systems are horizontal or face split coils, vertical or row split coils, and intertwined coil circuiting (Figure A-18).

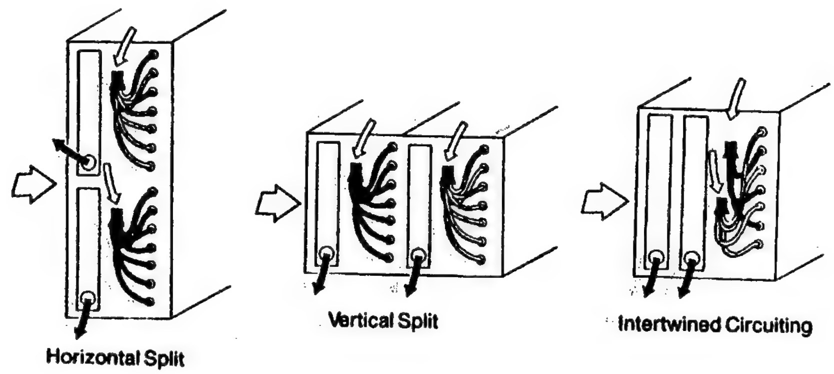


Figure A-18. Split Systems.

Source: *ASHRAE HVAC Systems and Equipment Handbook, 1992.* Used with permission.

Horizontal split/face split coils. Face splits are preferred for VAV applications as adequate superheat is assured for each coil section. This is true even at part load because all coil sections receive unconditioned air.

When the system changes to handle part load conditions, the mass flow rates decrease, and a solenoid valve upstream of one of the thermal expansion valves is closed. All the refrigerant now flows through the remaining distributor. With the horizontal split coils, the mass flow at the open distributor doubles, and it singly maintains loading above the minimum MBh/circuit. The inactive coil section bypasses unconditioned air that may cause problems for systems that use 100 percent outside air at part load. Face split coils are not recommended for 100 percent outside air applications in humid climates. At part load, this type of

* The minimum mass flow rate at which the distributor can provide uniform distribution to all circuits is expressed as MBh/circuit, and is a function of suction pressure. The maximum MBh/circuit is a function of distributor geometry and suction pressure. Stable performance is assured only when full load and part load MBh/circuit remains within this allowable range.

split coil can pose a potential problem because the system controllers try to keep the coil leaving air temperature constant. With a major part of the coil inactive, the active coil section must cool air to very low temperatures to maintain 55 °F leaving air. For example, if entering air is at 80 °F and half the coil is inactive, the active coil section must cool to 30 °F to achieve an average discharge of 55 °F. These conditions are conducive to coil frosting, and liquid refrigerant being returned to the compressor.

Vertical/row split coils. This type of split coils is not recommended for VAV applications. Compressor cycling can occur in row split coils. This is not bad in itself because reciprocating compressors can be used and will tolerate the cycling. However, compressor cycling upsets superheat control. Excessive superheat hampers compressor motor and discharge valve cooling. In the absence of superheat, liquid refrigerant is returned to the compressor. Reciprocating compressors are designed to tolerate brief periods of liquid in the suction line, but the combination of reduced airflow and humidity, and sustained initial temperature difference causes the upstream two row coil to produce a colder than anticipated leaving air temperature. The colder temperature leaving the upstream coil can hamper the ability of the downstream coil to provide adequate superheat. This superheat loss can occur for an extended period of time. If the loss of superheat lasts longer than the cycle rate of the VAV discharge air temperature controller, the compressor will likely fail.

Intertwined coil circuiting. The pitfalls of row and face split coils can be avoided by using intertwined coil circuiting. It provides more active fin surface at part load, and improved superheat capabilities at all load conditions.

At part load, the coil behaves like a coil with substantially greater fin surface, but without the penalty of higher airside pressure drop. By increasing the active fin surface at part load, the potential for coil frosting is reduced while maintaining excellent dehumidification. Superheat is not lost at part load conditions, and stabilizes quickly after a change in compressor or capacity.

Intertwined coil circuiting may require additional distributors and thermal expansion valves in some circumstances, but the DX/VAV stability at part load is worth the additions. Intertwined coils are best for almost all DX/VAV split system applications. They have been used extensively in packaged unitary equipment including rooftop and self-contained air conditioners in VAV applications.

If intertwined coils are not suitable, face split coils are acceptable if used with some type of supply air reset at part load. The row split should be avoided in VAV applications, but they are preferred in 100 percent outside air applications.

Filters

Filters are important for providing a comfortable and healthy air supply to the occupants, reducing dust deposits on room surfaces, and keeping interiors of HVAC system components clean. Filters and other air cleaning devices are available in four general types for four general purposes: (1) typical commercial filters to remove visible particles of dust, dirt, lint, and soot, (2) electrostatic filters to remove microscopic particles such as smoke and haze, (3) activated charcoal to destroy odors, and (4) ultraviolet lamps or chemicals to kill bacteria.

Both throwaway and cleanable filters are available. Throwaway filters are generally standard on smaller AHUs (less than 10,000 cfm). The standard commercial grade filters remove about 75 to 85 percent of the particles in the air. In hospitals and laboratories where a high degree of cleanliness is called for, high-efficiency filters are used.

Three different physical arrangements for filters in air handlers are flat, offset, and V-bank. The latter two provide more filter face area and, therefore, a lower face velocity across the filter. The maximum allowable face velocity for throwaway filters is 300 fpm versus a maximum of 500 fpm for cooling coils and 800 fpm for heating coils.

Filter banks may contain many throwaway filters that slide into the filter section channels on the top and bottom of each row of filters. The easiest way to change filters in large systems is to open access doors on each end of the filter bank. New filters are pushed into one end, while the used filters fall onto the floor at the other end. When the filter bank is accessible from only one end, a strip is used in the bottom channel. As the strip is pulled out, the farthest filter from the access door is pulled, pushing all the other filters in that line ahead of it (Figure A-19).

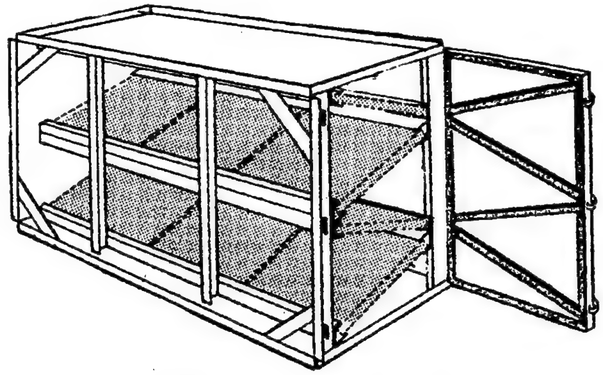


Figure A-19. Changing Filters.

Source: NAFA Guide to Air Filtration, 1993. Used with permission.

When dust loadings are expected to be quite heavy, the labor cost to continually replace filters can become prohibitive. Automatic filter changing can be done by using a roll filter (Figure A-20). The filter is advanced a few inches at a time, exposing new filter media at one end and rolling up dirty media at the other end. The advance of the filter is based on either a timer or a pressure-drop reading across the media. The latter is better because it exposes new media based on how much dirt the existing media has collected rather than on how long it has been in place.

For very critical jobs, a bag filter (Figure A-21) provides an extremely high cloth area, allowing the air to move through the filtering media very slowly. These are sometimes referred to as HEPA filters, which stands for high-efficiency particulate arrest. They are expensive to replace, and should be used with a less expensive throwaway filter upstream to filter out the larger size particulates.

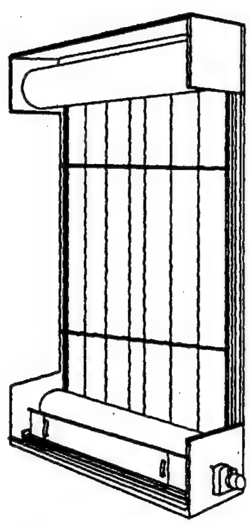


Figure A-20. Roll Filter Exposes Clean Media While Rolling Up the Dirty Media.

Source: NAFA Guide to Air Filtration, 1993. Used with permission.

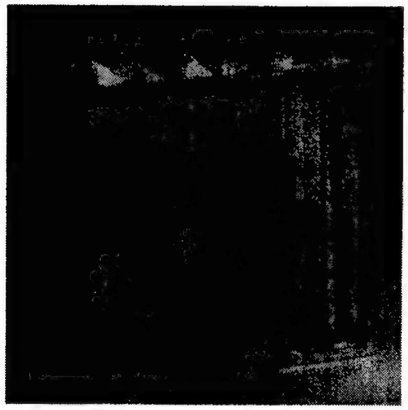


Figure A-21. Bag Filter Provides High Efficiency by Using Low Velocity Through the Filter Media.

Source: NAFA Guide to Air Filtration, 1993. Used with permission.

Mixing Box

A mixing box section (Figure A-22) is a convenient way to bring return air and outside air into the air handler.

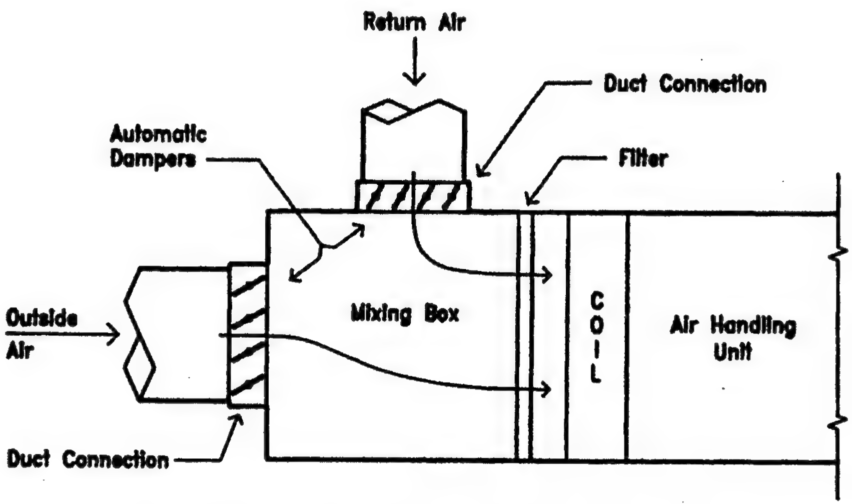


Figure A-22. Mixing Box for Return Air and Outside Air.

Source: Colen 1990. Used with permission of R.S. Means Company, Inc.

A damper is provided for each air stream to allow the controls technician to balance the percentage of outside air versus return air. The dampers may be either parallel blade or opposed blade (Figure A-23). Parallel blade damper sections are less expensive. Opposed blade damper sections provide better control characteristics.

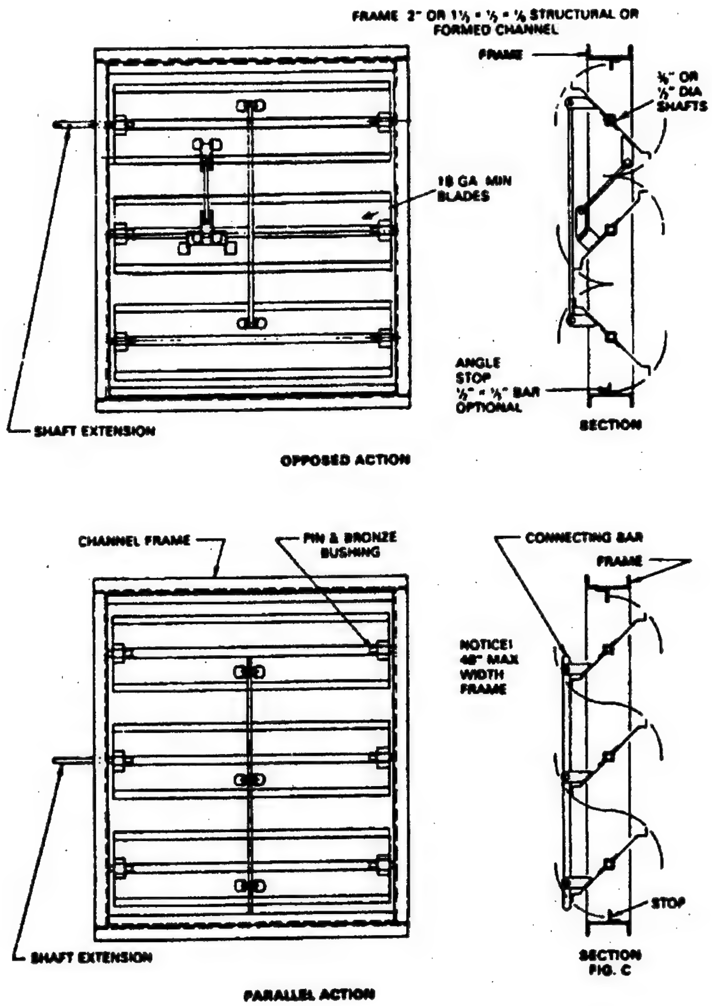


Figure A-23. Opposed and Parallel Blade Dampers.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

Humidification

Air leaving the central system of a VAV system is usually at 55 °F. Placing a humidifier in the air handling unit just before the cooling coils in a draw through setup would defeat its purposes for humidification. The steam would condense and drip, causing puddles in the AHU. If it were a rooftop system, this could lead to more serious problems.

Most humidifiers should be placed at least 10 ft from the AHU, if even this close. Since the VAV system has cool air and uses VAV boxes that are obstructions, it is best to place a humidifier in the duct after these boxes.

Humidification can be accomplished by direct injection of steam into the air stream, vaporizing water from a pan by heating it, passing air through a moist porous pad, or by spraying water from a nozzle into the air stream.

An example of a humidifier is the single-tube or Mini-Bank* multi-tube humidifier. These are specifically designed for application in hospital surgery rooms, intensive care units, delivery rooms, clean rooms, and where rapid steam absorption (in cool air) is required. If large ducts are used, the Maxi-Bank* may be used as it has an instantaneous total absorption within three feet of the tube bank, in any air temperature, and up to 50 percent relative humidity.

For a more detailed discussion on humidification, please refer to the Appendix A Annex.

---

* A commercial product of the "DRI STEEM" Humidifier Company, Hopkins, Minnesota.

3 Duct Design and Construction

Of the total cost of owning and operating a typical commercial building, maintenance and operation account for approximately 30 percent, while initial construction costs account for approximately 20 percent (SMACNA 1981). Some of the variables responsible for initial system cost are duct system material, system operating pressure, duct size and complexity, fan horsepower, sound attenuation, and space requirements. Of these, a major contributor to total building HVAC system annual energy cost is the energy demand of the fan distribution system. The fan energy cost can be minimized by reducing duct velocities and static pressure losses. This has a direct bearing on the system first cost, however, and could influence building cost as a result of using larger ductwork and equipment. It may also require more space and larger mechanical rooms.

Good duct design can result in first cost savings, substantial operating cost savings, and lower noise levels by minimizing and equalizing static pressure throughout the system. If the duct design is poor, additional costs may arise because designers tend to use pressure independent terminal controls and multiple point duct air pressure and airflow sensors for fan volume control, and extra controls may be needed to provide adequate stability.

Several general design and construction principles can be followed to control first costs and future operating costs:

- Use the minimum number of fittings possible. Most fittings are fabricated by hand, and the cost can be four to eight times that of straight section of duct.
- Consider the use of semi-extended plenums.
- Consider sealing ductwork to minimize air leakage. Sealing may allow equipment and ductwork sizes to be reduced.
- Use round duct where space allows, as round ductwork gives the lowest possible friction loss for a given perimeter.
- Maintain a rectangular duct aspect ratio as close to 1 to 1 as possible to minimize friction loss and cost.

The aspect ratio example in Table A-2 (based on 5000 cfm and 100 ft of duct) shows friction rates change as duct dimensions vary. The graphs in Figures A-24 and A-25 illustrate the installed and operating costs vs. aspect ratio.

Duct Dimensions		Duct Area			Metal Thickness			Duct Weight*	
Inches	Millimetres	Square Inches	Square Metres	Aspect Ratio	Gauge	Inches	Millimetres	Pounds per Foot	Kilograms per Metre
24 (diam.)	600 (diam.)	452	0.28	—	26	0.022	0.55	5.70	8.35
22 × 22	550 × 550	484	0.30	1 : 1	26	0.022	0.55	6.64	9.73
30 × 16	750 × 400	480	0.30	1.9 : 1	26	0.022	0.55	6.95	10.71
44 × 12	1100 × 300	528	0.33	3.7 : 1	22	0.034	0.85	13.12	19.21
60 × 10	1500 × 250	600	0.38	6 : 1	20	0.040	1.00	19.32	28.28
80 × 8	2000 × 200	640	0.40	10 : 1	18	0.052	1.31	31.62	46.29

*Duct Weight Based on 2 in.w.g. (500 Pa) Pressure Classification, 4 foot (1.22 m) Reinforcement Spacing. (Weight of Reinforcement and Hanger Materials **Not** Included.)

Table A-2. Friction Rate Vs. Aspect Ratio.

SMACNA *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

Duct Pressure Classification

The fan in an air distribution system provides the pressure to the airstream to overcome the resistance to flow of the fan itself, ductwork, air control dampers, cooling coils, heating coils, filters, diffusers, sound attenuation equipment, turning vanes, etc. These various components, duct surface friction, and changes in airflow direction or velocity cause various changes in pressure to occur in the duct system.

Figure A-26 illustrates typical pressure changes that may occur in a duct system. At any cross-section, the total pressure (TP) is the sum of the static pressure (SP) and the velocity pressure (VP). For all constant-area straight duct sections, the static pressure losses are equivalent to the total pressure losses. The pressure losses in the straight duct sections are called friction losses. The pressure losses increase more rapidly in the smaller cross-sectional area ducts. When duct cross-sectional areas are reduced abruptly (such as at B) or gradually (such as at F), both the velocity and velocity pressure increase in the direction of airflow. The absolute values of both the total pressure and static pressure decreases. The pressure losses at these points are dynamic pressure losses. From point D to E, there is a large jump in static pressure. As mentioned above, SP and TP in straight duct sections will increase or decrease with equal magnitude. Notice also the level of TP and SP at the fan is equal to atmospheric pressure. This jump in TP and SP is a result of the change from a negative pressure (on inlet side of fan) to a positive pressure (on the discharge of the fan). Increases in the

duct cross-sectional areas (at C and G) cause a decrease in velocity and velocity pressure, a continuing decrease in total pressure, and an increase in static pressure caused by the conversion of velocity pressure to static pressure. This increase in static pressure is commonly known as static regain.

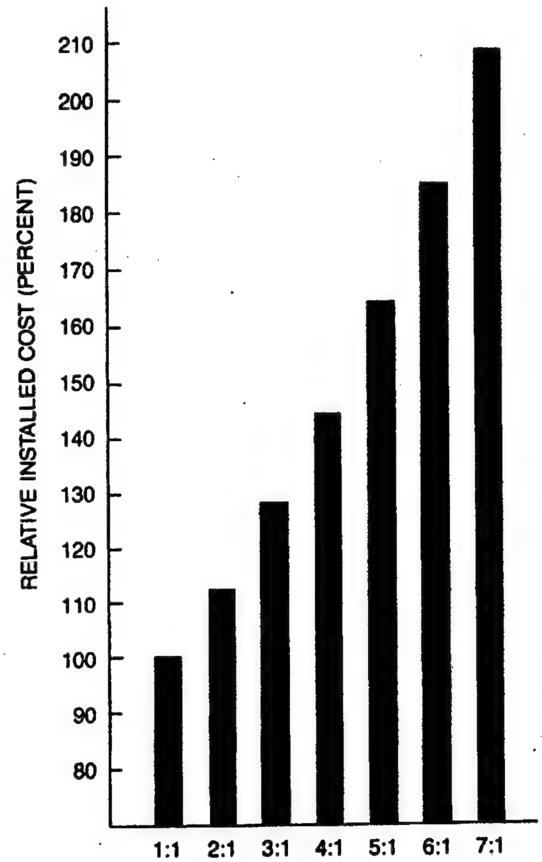


Figure A-24. Relative Installed Cost Vs. Aspect Ratio.

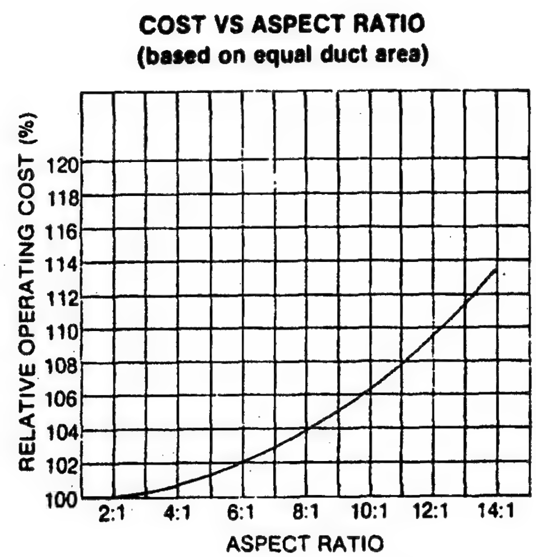


Figure A-25. Relative Operating Cost Vs. Aspect Ratio.

SMACNA *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

All changes in the ductwork static pressure should be indicated on the drawings. In VAV systems, the pressure classification for the ductwork from the primary AHU to the mixing boxes is based on Table A-3 (SMACNA 1995).

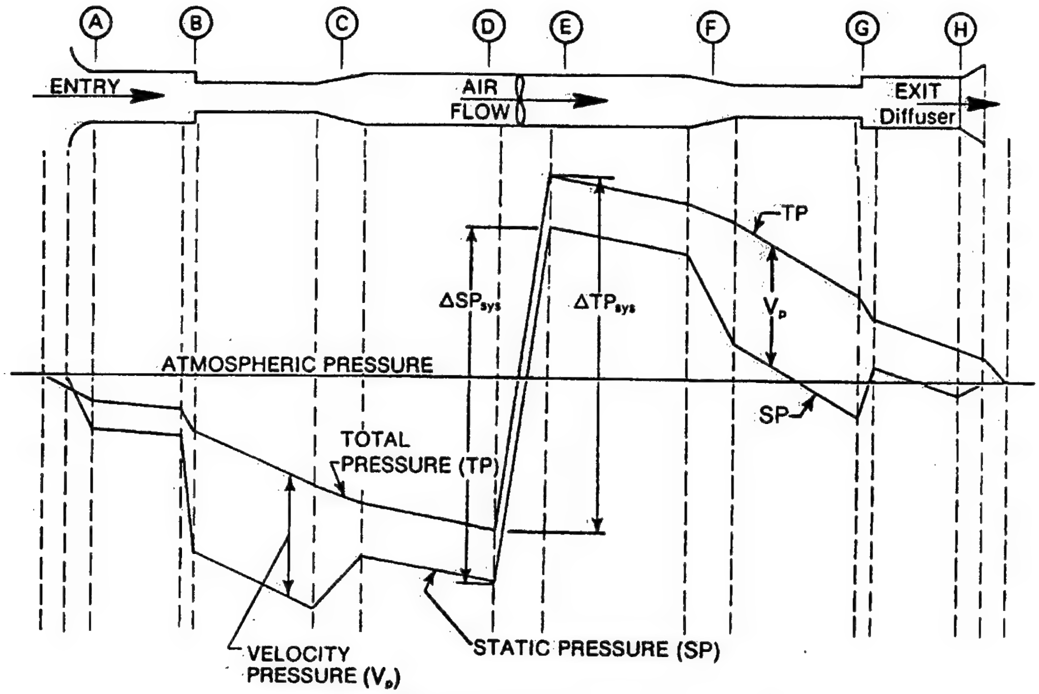


Figure A-26. Typical Duct System Pressure Changes.

SMACNA, *HVAC Systems–Testing, Adjusting and Balance*, 2nd Ed., 1993. Used with permission.

TABLE 1-1
PRESSURE CLASSIFICATION
FOR DUCTWORK

STATIC PRESSURE CLASS [INCHES (Pa) W.G.]	+1/2"	-1/2"	+1"	-1"	+2"	-2"	+3"	-3"	+4"	-4"	+6"	-6"	+10"	-10"
Rectangular Style	A	A	STD	STD	STV	A	A	A	A	A	A	A	A	A
Round Style					STV	STD			A	A			A	A
Flat Oval Style			STD		STV		A		A		A		A	
Flexible Style	A	A	STD	STD	STV		A		A		A		A	

Table A-3. Pressure-Velocity Classification for Ductwork.

SMACNA *HVAC Duct Construction Standards—Metal and Flexible*, 2nd Ed., 1995. Used with permission.

General Approach to Duct Design

After room loads and air quantities have been determined, the next step is duct design. The type of duct system needed is determined based on an economic analysis of the building design and use.

The two main categories of air systems are single duct and dual duct. The single duct system uses a single path duct distribution system with a common (variable) air temperature to feed all terminal apparatus, or it blends air from hot

and cold sources within the AHU and uses a single duct to feed each zone. Dual duct systems use a separate cold and warm air duct distribution system that blends the air at the terminal apparatus.

After determining the type of duct system needed, duct design proceeds in the following order:

1. Locate air outlets and select size and type required for proper air distribution.
2. Locate return and exhaust air devices.
3. Select straightening vanes and dampers to be used with outlet devices to provide uniform face velocity and balancing.
4. Determine number of perimeter and interior zones.
5. Draw a preliminary schematic diagram for the ductwork that will provide the most efficient and economic path. Indicate the design airflows throughout the system.

Locating and Selecting Outlets

The number one concern in evaluating the air distribution in a space is the comfort of the occupants. The normal air velocity used for comfortable air distribution is 50 fpm while the acceptable range is from 25 to 75 fpm.

Outlets should be located to distribute the air as uniformly as possible throughout the room. Stagnant air is eliminated by an effective use of entrainment, which is the process by which the velocity of the air discharged from an outlet induces movement of the air already present in the room and blends the two.

In selecting outlets, keep in mind that cool air tends to drop away from the outlet, and warm air tends to rise to the ceiling. Select air outlets that provide acceptable air distribution for both cool and warm air.

The following order is followed for supply outlet location and selection:

1. Determine room supply air quantity from heating and cooling load calculations and design ventilation requirements.
2. Select type and quantity of outlets for each room and evaluate:
   a. Outlet cfm
   b. Outlet throw pattern
   c. Building structural characteristics

   d. Aesthetic architectural requirements
   e. Integration with other building systems, i.e. lighting, ceiling grids, partitions, etc.
3. Locate outlets to provide uniform room temperature using as uniform an air distribution pattern as possible.
4. Select proper outlet size from manufacturer's catalog data considering:
   a. Outlet cfm
   b. Discharge velocity (throw)
   c. Distribution pattern
   d. Total pressure loss
   e. Sound level.

Tables A-4 through A-6 provide a general guide for the proper selection of outlets based on design requirements of cfm per square foot and air changes per hour (SMACNA 1990).

Type of Outlet	Floor Space: CFM/per Sq. Foot	Floor Space: l/s per Sq. Metre	Approximate Maximum Air Changes/Hour For 10 Ft. (3 m) Ceiling Height
Grilles & Registers	0.6 to 1.2	3 to 6	7
Slot Diffusers	0.8 to 2.0	4 to 10	12
Perforated Panel	0.9 to 3.0	5 to 15	18
Ceiling Diffuser	0.9 to 5.0	5 to 25	30
Perforated Ceiling	1.0 to 10.0	5 to 50	60

Table A-4. General Guide for Selecting Supply Outlet Type.

SMACNA *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

Group—Type	Mounting	Discharge Direction	Characteristics	
			Cooling	Heating
A High Sidewall Grilles Sidewall Diffusers Ceiling Diffusers Slot Diffusers (Parallel Flow)	Ceiling, High Sidewall	Horizontal	Good mixing with warm room air. Minimum temperature variation within room. Particularly suited to cooling applications	Large amount of stagnant air near floor. In interior zones where loading is not severe, stagnant air is practically non-existent
Variable Area Grille Variable Area Diffuser	Ceiling, High Sidewall	Horizontal Specially adapted for variable volume systems	Maintain design air distribution characteristics as air volume changes	Maintain design air distribution characteristics as air volume changes
B Floor Grilles Baseboard Units Fixed Bar Grilles Linear Grilles	Floor, Low Sidewall, Sill	Vertical Non Spreading Air Jet	Small amount of stagnant air generally above occupied zone	Smaller amount of stagnant air than Group A outlets
C Floor Grilles Adjustable Bar Grilles Linear Diffusers	Floor, Low Sidewall, Sill	Vertical Spreading Air Jet	Larger amount of stagnant air than Group B outlets	Smaller amount of stagnant air than group B outlets—particularly suited to heating applications
D Baseboard Units Grilles	Floor, Low Sidewall	Horizontal	Large amount of stagnant air above floor in occupied zone—recommended for comfort cooling	Uniform temperature throughout area. Recommended for process applications
E Ceiling Diffusers Linear Grilles Grilles Slot Diffusers (Vertical Flow) Sidewall Diffusers	Ceiling, High Sidewall	Vertical	Small amount of stagnant air near ceiling. Select for cooling only applications.	Good air distribution. Select for heating only applications

Table A-5. Supply Air Outlet Performance.

SMACNA *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

Locating and Selecting Returns and Exhaust Devices

Return air inlets are generally located so that the room air returned is the greatest temperature difference from that being supplied to the room. Air motion is not significantly affected by the location of return and exhaust inlets. Also, the location of return and exhaust inlets will not compensate for ineffective supply air distribution. A return air inlet that is located directly in the primary airstream of the supply outlet will short circuit the supply air back into the return without mixing with room air.

Special situations that require careful attention by the designer are the location of return and exhaust inlets in bars, kitchens, lavatories, dining rooms, club

Type	Characteristics	Applications
Fixed blade grille	Single set of vertical or horizontal blades	Long perimeter grille installations
Adjustable single deflection blade grille	Single set of vertical or horizontal adjustable blades	Sidewall installation where single plane air deflection is required
Adjustable double deflection blade grille	One set of vertical and one set of horizontal adjustable blades	Preferred grille for sidewall installation provides both horizontal and vertical air deflection
Stamped plate grilles	Stamped from single sheet of metal with square, round or ornamental designed openings	No adjustment of air deflection possible. Use for architectural design purposes only
Variable area grille	Similar to adjustable double deflection blade grille with means to effectively vary the discharge area	Use with variable volume system to minimize variation of throw with variable supply air volume
Curved blade grilles	Curved blades to provide horizontal air pattern	Ceiling installation High sidewall installation Perimeter installation
Perpendicular-flow slot diffuser	Generally 25 to 1 dimensional aspect ratio with maximum height of 3 inches (75 mm)	High sidewall installation Perimeter installation in sills, curbs and floors
Parallel-flow slot diffuser	Generally 25 to 1 dimensional aspect ratio with maximum height of 3 inches (75 mm)	Ceiling installation
Air light fixture slot diffuser	Use in conjunction with recessed fluorescent light fixtures with fixed or adjustable air discharge patterns	Ceiling installation— Order to match light fixture
Multi-passage round ceiling diffuser	Series of flaring rings or louvers forming series of concentric air passages	Install in center of area served
Multi-passage square and rectangular ceiling diffuser	Series of flaring rings or louvers forming series of concentric air passages	Install in center of area served
Adjustable pattern round ceiling diffuser	Series of flaring rings or louvers forming series of concentric air passages. Air discharge pattern adjustable from horizontal to vertical or down blow pattern	Install for control of diffuser discharge pattern or where specific requirement to direct airflow pattern either horizontal or vertical.
Adjustable pattern square and rectangular ceiling diffuser	Series of flaring rings or louvers forming series of concentric air passages. Air discharge pattern adjustable from horizontal to vertical or down blow pattern	Install for control of diffuser discharge pattern or where specific requirement to direct airflow pattern either horizontal or vertical.
Multi-pattern square and rectangular ceiling diffuser	Special louvers discharge air in one or more directions	Install in center of area served or adjacent to partitions. Set pattern according to flow requirements.
Half round diffuser	Matches round diffuser	Install in ceiling adjacent to partition or high sidewall
Supply and return concentric diffuser	Combination diffuser with return grille in center of diffuser	Install in center of area served
Light fixture air diffuser combination	Combination diffuser-light fixture	Ceiling installation combined with light fixture pattern
Perforated face diffuser	Perforated face plate with or without deflection device to obtain a horizontal discharge pattern	Install in center of area served or control discharge pattern when installed off center of area served
Variable area diffuser	Parallel or concentric passages or perforated face with means to vary discharge area	Use with variable volume system to minimize variation to throw with variable supply air volume
Air distributing ceilings	Ceiling system provided with round holes or slots	Use with ceiling supply plenum—particularly suited to large zones of uniform room temperature
Linear grille	Linear slot width ½ to 1 inch (12 to 25 mm), continuous length with adjustable airflow blades	Ceiling and perimeter with air deflection adjustable from 1-way horizontal to vertical to 2-way horizontal
Egg crate grille	Fixed square grid	Ceiling or sidewall (no pattern adjustment)
High capacity double deflection blade grille	One set vertical and one set horizontal adjustable blades. Blades are deep & wide spaced	High sidewall installation where high capacity and low discharge velocity are required
Drum louvers and adjustable high capacity diffusers	Adjustable direction core	High sidewall or ceiling installation, where directional and/or long throw required provides spot heating or spot cooling to areas of high load requirements

Table A-6. Supply Air Outlet Types.

SMACNA *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

rooms, etc. These normally should be located near or at the ceiling level to collect the warm air "build-up," odors, smoke, and fumes.

Generally, the procedure for inlet selection and location is as follows:

1. Determine from design load calculations the quantity of room return air and the quantity of exhaust air.
2. Taking into consideration the following, select the type and quantity of inlets for each room.
   a. Inlet cfm
   b. Inlet velocity
   c. Architectural requirements
3. Locate inlets to enhance room air circulation, and to remove undesirable air.
4. Using manufacturer's catalogs, select proper inlet size considering:
   a. Inlet cfm
   b. Inlet velocity
   c. Total pressure loss
   d. Sound level.

Zoning

If the designer is not given the number of different zones of temperature control, then he/she must determine the number of both perimeter zones and interior zones. Generally, the exterior zone will be divided into zones that will be determined by building exposure (north, east, south, or west exposure). These perimeter zones can be further subdivided into smaller control zones, depending on variations in internal load or a requirement for individual occupant control, such as a computer room, a conference room, or a private executive office. Likewise, the interior zones may also be broken down into control zones to satisfy individual office requirements or variations created by internal loads such as lights, people, or machinery.

Preliminary Duct Layout

After determining the zones for the building, a preliminary layout of the ductwork will need to be drawn. This diagram should indicate the design airflows throughout the system, and illustrate the most efficient and economic paths to the selected zones and outlets. It is suggested that this preliminary layout be drawn on a tracing of the architectural floor plans. This will help the designer coordinate the ductwork with the structural limitations of the building and other

services for the building. It will also enable the designer to have a good representation of the relationship of air terminals, branch ducts, main ducts, risers, etc.

Duct Sizing

Ductwork for HVAC duct systems usually is sized as round ductwork first. Then, if rectangular ducts are desired, duct sizes are selected to provide flow rates equivalent to those of the round ducts originally selected. If the rectangular duct sizes are calculated directly from the actual duct cross-sectional area, the resulting duct sizes will be smaller, causing a greater velocity and larger friction loss. The efficiency of duct fabricating machinery that reduces shop labor cost has encouraged a trend toward returning to the use of rectangular ductwork for high velocity systems. A disadvantage can be noise generated by turning vanes and balancing dampers. Most knowledgeable designers still feel that round spiral ducts can be used more expediently and may have more overall advantages.

Using one of the design methods discussed in the next section, the designer must select duct sizes and then calculate total pressure or static pressure losses. He/she will then need to determine if the ductwork will fit into the building. Consideration must be given to:

1. Additional space required beyond bare sheet metal sizes for reinforcing, circumferential joints, and hangers.
2. External insulation or duct liner.
3. Clearance for piping, conduit, light fixtures, etc.
4. Clearance for removal of ceiling tile.
5. Space requirement for air terminals, mixing boxes, pressure reducing boxes, fire and smoke dampers, reheat coils, etc.

Duct Design Methods

Of several different design methods, the most universally used are the equal friction and the static regain methods. The following objectives and considerations should be adhered to for all methods:

- Design the duct system for the most efficient airflow from the fan to the terminal devices.
- Consider energy conservation in the fan selection, duct configuration, duct wall heat gain or loss, etc.
- Consider sound attenuation.

- Indicate the location of all life safety devices such as fire dampers, smoke dampers, etc.
- Make provisions for testing, adjusting, and balancing.
- Consider the pressure losses that occur from tie rods and other duct obstructions.
- Consider that round ducts are preferred on higher pressure systems due to their lesser construction costs.

Equal Friction Method

Probably the most widely used method of duct sizing is the equal friction method. It is used for sizing low pressure supply air, return air, and exhaust air systems. It is also being adapted by many for use in medium pressure systems.

Advantages:

- Requires less balancing for symmetrical layouts.
- Automatically reduces air velocities in the direction of airflow. This reduction will decrease the chances of introducing airflow generated noise from high velocities.
- System velocity may be readily checked at any point.
- Disadvantages:
- No natural provision for equalizing pressure drops in the branches.
- No means of providing the same static pressure behind each supply or return terminal device.

Procedure:

1. Select initial velocity in main duct near fan.
2. Using selected velocity and design cfm, determine the friction rate.
3. This same friction rate is then maintained throughout the system, and an equivalent round duct is calculated.
4. Select rectangular duct sizes if needed.
5. Total friction loss in the duct system is calculated for the duct run having the highest resistance.

Static Regain Method

The static regain method of duct sizing is a theoretically sound method, and meets the requirements of maintaining uniform static pressure at all branches and outlets. It is normally not applicable for return air systems where the air-

flow is towards the fan. It is more complex to use than the equal friction method. The static regain method of duct sizing may be used to design supply air systems of any velocity or pressure.

The basic principle of the static regain method is to size a duct run so that the increase in static pressure (regain due to reduction in velocity) at each branch or air terminal just offsets the friction loss in the succeeding section of duct. The static pressure will then be the same before each terminal and at each branch.

Advantages:

- Duct velocities are systematically reduced, allowing a large portion of the velocity pressure to convert to static pressure, which offsets the friction loss in the succeeding section of duct. According to the SMACNA's HVAC Systems--Duct Design manual, the static regain, which is assumed at 75 percent for the average duct system, could be as high as 90 percent under ideal conditions.
- Duct system will stay in balance because the losses and gains are proportional to a function of the velocities. Therefore, it is an excellent method for designing variable air volume systems.

Disadvantage:

- Oversized ducts can occur at the ends of long branches.

T-Method

The T-method is an optimization procedure that minimizes an objective function. This method was named after its inventor, Tsal et al. (1988). It is based on the same tee-staging idea as dynamic programming. However, phase level vector tracing is eliminated by optimizing locally at each stage. This modification reduces the number of calculations, but usually requires three iterations.

The T-method comprises the following major procedures:

1. *System Condensing.* This procedure condenses a branch Tee system into a single imaginary duct section with identical hydraulic characteristics and the same owning cost as the entire system. By using

$$K_{1-3} = [(K_1 + K_2)^{0.833}]^{1.2}$$

two children sections and a parent section may be replaced by one condensed section (ASHRAE 1989). By applying this equation from junction to junction in the direction of the root section fan, the entire supply and return systems can be condensed into one section.

2. *Fan Selection.* From the condensing system, the ideal optimum fan total pressure P1(opt.) is calculated and used to select a fan. If a fan with a different pressure is selected, its pressure (P-opt.) is considered optimum.

3. *System Expansion.* The expansion process distributes the available fan pressure P(opt.) throughout the system. Unlike the condensing procedure, the expansion procedure starts at the root section and continues in the direction of the terminals. More information on this method may be found in the 1997 ASHRAE Fundamentals Handbook. This method is one of the best ways to design ducts economically.

Extended or Semi-extended Plenums

The use of an extended or semi-extended plenum is not an entirely different method of duct or system sizing, but a combination of good design and cost saving ideas using conventional duct sizing techniques.

An extended plenum is a trunk duct of constant size, usually at the discharge of a fan, fan-coil, mixing box, constant volume box, etc., extended as a plenum to serve multiple outlets and/or branch ducts. A semi-extended plenum system is similar to extended plenum design, and incorporates a minimum number of size reductions due to decreasing volume.

Velocity Reduction

This method should only be used by experienced designers. A system velocity is selected at a section next to the fan, and arbitrary reductions in velocity are made after each branch or outlet. Balancing is attempted mainly by use of good dampers at strategic locations.

Total Pressure

A refinement of the static regain method that allows the designer to determine the actual friction and dynamic losses at each section of the duct system.

Constant Velocity

This method is better adapted to high pressure systems where attenuated terminal boxes are used to reduce the velocity and noise before distribution of air to occupied spaces.

Duct Fabrication and Construction

Duct walls, transverse joints, and reinforcements at or between joints and supports make up the basic elements of duct construction. These elements form an integrated combination for each pressure class (Table A-3) and duct size. Each size in a pressure class has a minimum duct wall thickness and a minimum specification for joints, reinforcements, etc. To construct a rectangular duct with the correct thickness, joints, and reinforcing, the first step is to determine the pressure class assigned to the duct by the designer. Then the appropriate reinforcement table from SMACNA's HVAC Duct Construction Standards is used (see Figure A-27). The greater of the duct's dimension is the one that is used to determine the gage of the sheet metal for all sides. Depending on the gage selected, the table also shows whether reinforcement will be required or not. The table also indicates the right type of joint connection to use and spacing intervals for joints or reinforcements.

Round ducts have a high strength to weight ration, uses the least material to convey air at a given friction loss, and is comparatively easy to seal. SMACNA specifies the right gage to use depending on the pressure, diameter of the duct and whether a spiral or longitudinal seam is used. A 10 in. w.g. negative pressure is the maximum that is allowed for round ducts.

Flat oval ducts combine the advantages of round and rectangular ducts because they may fit in spaces where there is not enough room for round ducts, and can be joined using round duct assembly techniques. Compared to a corresponding size of rectangular duct, a flat oval duct has much less flat surface that is susceptible to vibration and requires less reinforcement. SMACNA specifies minimum duct wall thickness dependent on the duct's major dimension width and whether spiral or longitudinal seams are used. The reinforcement size and spacing interval is the same as that specified for rectangular ducts. However, flat oval ducts are for positive pressure applications only.

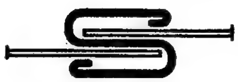
READING GUIDE SUMMARY

Example: 54" x 18" duct, 5 ft joint spacing. On 54" sides use F joints on 22 ga. On 18" sides flat slips or drives qualify per column 2.
Example: 54" x 30" duct, 22 gage. Use F at 5 ft. on 54". On 30" use D at 5 ft. or E at 10 ft. If you put joints on the 30" side at 5 ft. spacing, they must be D rated.

Comment: If the table requires a letter code, all joints on that side must qualify for the minimum code letter related to the minimum gage and the spacing.

Use Drive Slip or Hemmed "S" Slip on duct gage in column 2


DRIVE SLIP
OR

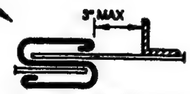

HEMMED "S" SLIP

Duct Gage	26 to 22	20	18	16
Minimum Flat Slip and Drive Gage	24	22	20	18

Joint Option: Backup member qualifies Hemmed "S" Slip - Reinforced or Drive Slip - Reinforced for letter code when selected from Table 1-10.

HEMMED "S" SLIP

OR



DRIVE SLIP - REINFORCED

Spacing refers to letter code: use joint-to-joint, joint-to-intermediate or intermediate-to-intermediate. Columns 3 to 10 are alternatives.

The drive slip is accepted as being A, B or C rated up to 20" length.

1" W.G. STATIC POS. OR NEG.

TABLE 1-4 RECTANGULAR DUCT REINFORCEMENT

DUCT DIMENSION	NO REINFORCEMENT REQUIRED	REINFORCEMENT CODE FOR DUCT GAGE NO. — REINFORCEMENT SPACING OPTIONS: 10'	8'	6'	5'	4'	3'	2 1/2'	2'
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
10"dn	26 ga.	NOT REQUIRED							
11, 12"	26 ga.								
13, 14"	24 ga.	B-26	B-26	B-26	B-26	B-26	A-26	A-26	A-26
15, 16"	22 ga.	B-24	B-26	B-26	B-26	B-26	B-26	B-26	A-26
17, 18"	22 ga.	B-24	B-26	B-26	B-26	B-26	B-26	B-26	B-26
19, 20"	20 ga.	C-24	C-26	C-26	C-26	C-26	B-26	B-26	B-26
21, 22"	18 ga.	C-24	C-24	C-26	C-26	C-26	B-26	B-26	B-26
23, 24"	18 ga.	C-24	C-24	C-26	C-26	C-26	C-26	B-26	B-26
25, 26"	18 ga.	D-22	D-24	C-26	C-26	C-26	C-26	C-26	B-26
27, 28"	16 ga.	D-22	D-24	D-26	C-26	C-26	C-26	C-26	C-26
29, 30"	16 ga.	E-22	D-24	D-26	D-26	C-26	C-26	C-26	C-26
31-36"	NOT DESIGNED	E-20	E-22	E-24	D-24	D-26	C-26	C-26	C-26
37-42"		F-18	F-20	E-24	E-24	E-26	D-26	D-26	C-26
43-48"		G-18	G-18	F-20	F-22	E-24	E-26	E-26	D-26
49-54"		H-18	H-18	G-20	F-22	F-24	E-24	E-24	E-24
55-60"			H-18	G-20	G-22	F-24	F-24	E-24	E-24
61-72"				H-18G	H-18G	H-22G	F-24	F-24	F-24
73-84"				I-18G	I-18G	I-20G	H-22G	H-22G	G-22
85-96"					I-18H	I-18H	I-20G	H-20G	H-22G
97-108"						I-18G	I-18G	I-18G	I-18G
109-120"							I-18H	I-18H	I-18G

See page 1-15. Circles in the Table denotes only column numbers. For column 2, see Fig. 1-7. For Columns 3 through 9, see Introduction to Schedules. The number in the box is minimum duct gage; the alphabet letter is the minimum reinforcement grade for joints and intermediates occurring at a maximum spacing interval in the column heading. A listing such as H18G means that the H may be downsized to G with a tie rod. At higher pressures and large widths, a reinforcement such as Jt means that only tie rodded members are given.

TABLE 1-11 (Option) TRANSVERSE JOINT

MINIMUM RIGIDITY CLASS	EI^*	T-10 STANDING S — H x T	WT/LF
A	0.5	Use B	
B	1.0	1 x 26 ga.	0.6
C	1.9	1 x 22 ga.	0.8
D	2.7	1 x 1/8 x 20 ga.	0.9
E	6.5	1 1/8 x 18 ga.	1.0
F	12.8	Use G	
G	15.8	1 5/8 x 18 ga.	1.3
H	22		
I	69		
J	80		
K	103		
L	207		

OR

TABLE 1-10 (Option) INTERMEDIATE

REINF. CLASS	EI^*	ANGLE — H x T (MIN)	WT/LF
A	0.5	Use C	
B	1.0	Use C	
C	1.9	C1 x 16 ga. C3/4 x 1/8	0.40 0.80
D	2.7	H3/4 x 1/8 C1 x 1/8	0.57 0.80
E	6.5	C1 1/4 x 12 ga. H 1 x 1/8	0.8 0.8
F	12.8	H 1 1/4 x 1/8	1.02
G	15.8	1 1/2 x 1/8	1.23
H	22 (+) 26.7 (-)	1 1/2 x 3/16 2 x 1/8	1.78 1.65
I	69	C2 x 3/16 2 1/2 x 1/8	2.44 2.1
J	80	H2 x 3/16 C2 x 1/4 2 1/2 x 1/8 (+)	3.2 2.1
K	103	2 1/2 x 3/16	3.1
L	207	H2 1/2 x 1/4	4.1

C angle is cold rolled.
H angle is hot rolled.

Figure A-27. Example SMACNA Duct Reinforcement Table.

SMACNA *HVAC Duct Construction Standards–Metal and Flexible*, 2nd Ed., 1995, p 1.14

Fibrous Glass Ductwork

Fibrous glass ducts are fabricated from sheets of materials that have been manufactured from resin bonded inert and inorganic glass fibers. A factory applied facing (typically aluminum or reinforced aluminum) is applied to one face, and serves as a finish and a vapor barrier. Fibrous glass air ducts have been limited to 2 in. w.g. pressure and below.

The characteristics and requirements of fibrous glass ducts are:

- Maximum static pressure in duct - 2 in. w.g. positive or negative
- Maximum air velocity - 2000 fpm
- Maximum allowable board deflection - Transverse span/100 ft
- Maximum allowable stress in steel reinforcement - 22,000 psi
- Moisture - Moisture absorption of the board will not exceed 2 percent by weight under conditions of 120 °F DB at 95 percent RH for 96-h duration.
- Board fatigue - No significant deformation or deficiency of duct section after 25,000 cycles at 5 cycles per minute from natural sag to span/100 deflection.
- Temperature - 250 °F maximum inside the duct. 150 °F maximum ambient outside the duct.

SMACNA has a complete publication on Fibrous Glass Duct Construction Standards. It is suggested that this manual be referenced when using fibrous glass ductwork.

Sealing Ducts

To effectively close joints and seams, duct construction must have good workmanship. Ducts that are sealed as described in Table A-7 are expected to have leakage less than 5 percent of the system operating airflow. If less leakage is desired, seal all transverse joints in Class C. Refer back to Table A-3 for duct pressure classifications.

The terms "seal" or "sealed" refers to use of mastic or mastic plus tape or gasketing, as appropriate. Liquids, mastics, gaskets, and tapes have all been used as sealants. Selecting the most appropriate sealant depends on joint configuration clearances, surface conditions, temperature, the direction of pressure, and pre-assembly or post-assembly placement. Tapes should not be applied to dry metal or to dry sealant. Foil tapes are not suitable. Liquids and mastics should be used in well ventilated areas, and the precautions of manufacturers followed. Oil based caulking and glazing compounds should not be used. Gasketing should

be made of material with long life and suitable for the service. Use of thermally actuated products or products of a single source nature is not prohibited for quality sealing. The variety of sealant materials and their performance characteristics are such that a uniform specification has not yet been assembled.

TABLE 1-2		
STANDARD DUCT SEALING REQUIREMENTS		
SEAL CLASS	**Sealing Requirements**	**Applicable Static Pressure Construction Class**
A	Class A: All Transverse joints, longitudinal seams, and duct wall penetrations	4" w.g. and up (1000 Pa)
B	Class B: All Transverse joints and longitudinal seams only	3" w.g. (750 Pa)
C	Class C: Transverse joints only	2" w.g. (500 Pa)
In addition to the above, any variable air volume system duct of 1" (250 Pa) and 1/2" w.g. (125 Pa) construction class that is upstream of the VAV boxes shall meet Seal Class C.		

Table A-7. Seal Classes for Ductwork.

SMACNA *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

Provisions During Design for Testing, Adjusting, and Balancing

Providing proper physical layouts for testing, adjusting, and balancing (TAB) the airflow in the system after the building is completed is a must. Sufficient lengths of straight duct must be provided in an accessible area to allow the TAB personnel to perform their function properly. TAB personnel must be able to determine the total system airflow with a reasonable degree of accuracy. This also applies in TAB work of the critical branches of the distribution system.

It is important for system air to get to the occupied space with minimal losses caused from leakage and resistance, with proper mixing of air, and without temperature changes from heat gains or losses. Also important are the noise, drafts, and efficiency with which air is delivered. The means to meet these requirements are the proper design of ductwork and outlets.

The designer should give special attention to the balancing and adjusting process during the design. It is necessary that the balancing capability be designed into the system initially. Below are some considerations to make when designing duct systems.

- Application of single blade, quadrant volume dampers just behind diffusers and grilles may tend to throw air to one side of the outlet, preventing uniform airflow across the outlet face or cones.

- A slight opening of an opposed blade volume damper will generate a relatively high noise level as the air passes through the damper opening under system pressure.

- To minimize generated duct noise at volume dampers, indicate damper locations at least two diameters from a fitting, and as far as possible from an outlet.

- All portions of the main return air duct system require manual balancing dampers at each duct inlet.

- Avoid placing a return air opening directly in or adjacent to the return air plenum without a noise attenuator. Lining of the duct behind the opening normally will not reduce the transmitted noise to acceptable levels.

- Mixing boxes should be located so the discharge ductwork will minimize air turbulence and stratification.

- Provide the necessary space around components of the duct system to allow a TAB technician to obtain proper readings. Allow straight duct sections of 7-1/2-in. duct diameters from fan outlets, elbows, or open duct ends for accurate traverse readings.

- Ductwork to and from air conditioning equipment should be designed carefully so stratified air may be mixed properly before entering branch ducts or equipment.

- Splitter-type dampers should be regarded as air diverters only, with maximum effectiveness when present on duct systems exhibiting low resistance to airflow.

- Manually operated, opposed blade or single blade, quadrant-type volume dampers should be installed in each branch duct takeoff after leaving the main duct to control the amount of air entering or leaving the branch.

- Turning vanes should be installed so air leaving the vanes is parallel to the downstream duct walls. Double thickness or single thickness extended edge turning vanes should be utilized in all rectangular elbows.

- Manual volume dampers should be provided in branch duct takeoffs to control the total air to the face dampers of the registers or diffusers. Use of extractors is not recommended because they can cause turbulence in the main trunk duct thereby increasing the system total pressure, and affecting the performance of other branch outlets downstream. Register or diffuser dampers cannot be used for reducing high air volumes without inducing objectionable air noise levels.

- Do not use extractors at branch or main duct takeoffs to provide volume control. Extractors are principally used to divert air to branch ducts.

- Adequate size access doors should be installed within a normal working distance of all volume dampers, fire dampers, pressure reducing valves, reheat coils, mixing boxes, blenders, constant volume regulators, etc. that require adjustments within the ductwork. Coordinate locations with the architect.

- Provide for test wells, plugged openings, etc., normally used in TAB procedures.

4 VAV Boxes, Diffusers, and Dampers

VAV Boxes

A VAV box (terminal, terminal unit, throttling unit) is a device that is located where the supply duct or duct branch terminates, and the air is introduced into the space to be conditioned. Its functions include supplying air at proper temperature, regulating airflow, reducing pressure, and attenuating noise that is generated within the unit. The following are descriptions of the various types of VAV boxes.

Fan Powered VAV Box

The fan powered VAV box (Figure A-28) induces free heat from lighting, people, and other equipment.

Single Duct VAV Terminal

This type has only one duct connection (Figure A-29), and is supplied with air at a temperature that will take care of the cooling load. It is used only in spaces where cooling only is required year round, and where the variation in load is relatively small.

Single Duct VAV Box with Reheat

Reheat capability is added through the use of electric coil, hot water coil, or steam. With a decrease in cooling load, the room thermostat resets the regulator to the minimum volume setting. With a further decrease in load, the thermostat opens the reheat valve.

Dual Duct VAV Terminal

The dual duct VAV terminal (Figure A-30) is supplied with both hot and cold air. The inlet valve is positioned by a pneumatic motor or electric motor in response to a room thermostat to supply air at the proper temperature to satisfy the load

within the space. With a decrease in cooling load, the room thermostat resets the regulator to the minimum volume setting. With a further decrease in load, the thermostat gradually opens the warm air damper.

Bypass VAV Terminal

This type of VAV box (Figure A-31) has a constant cfm input. Unneeded air is bypassed directly into the ceiling plenum. Bypassing of the excess air out of the system and into the plenum will maintain the proper airflow into the occupied space and help ensure the comfort of the occupants.

Retrofit Terminals

Existing constant volume systems can be converted to VAV systems for the purpose of conserving energy (Figure A-32). Low velocity constant volume reheat, high velocity constant volume reheat systems, and the double duct constant volume system can be converted by using an appropriate retrofit terminal.

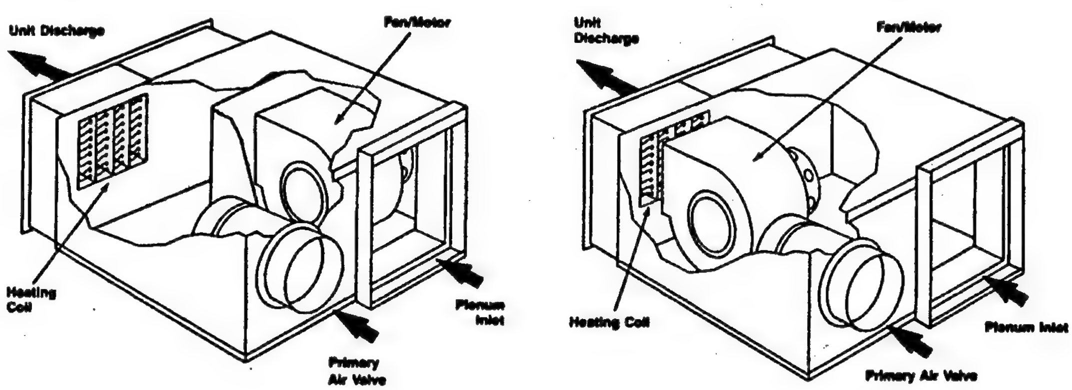


Figure A-28. Fan Powered VAV Box.
Provided by Titus.

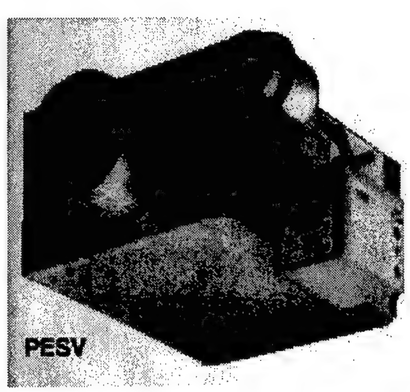


Figure A-29. Model ESV Single Duct VAV Terminal.
Provided by Titus.

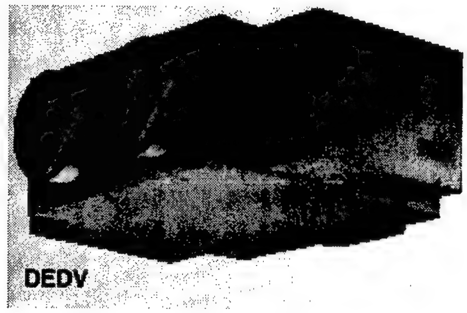


Figure A-30. Model MDV/MDC Dual Duct Terminal.

Provided by Titus.

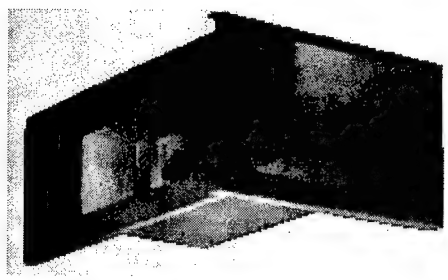


Figure A-31. Titus Model ZQCV Bypass Terminal.

Provided by Titus.

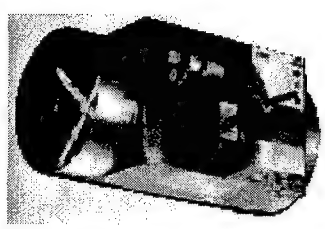
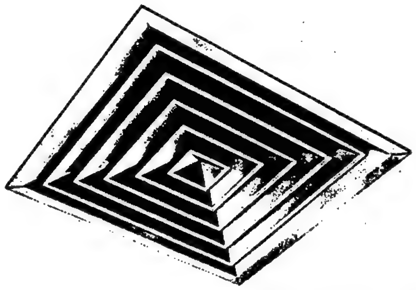


Figure A-32. VAV Retrofit Terminal.

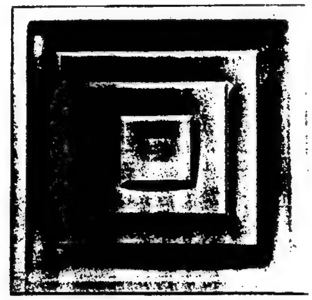
Provided by Titus.

Diffusers

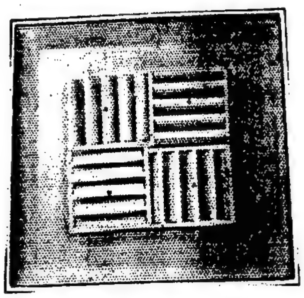
Air diffusers discharge supply air. Diffusers come in many different shapes and sizes and are usually located in the ceiling. Deflecting members in the diffuser discharge the supply air in various directions to promote mixing of primary air with secondary room air. Diffusers can be selected based on general configuration, sound level, supply or return, throw, airflow rate, balancing devices, and pressure drop. Figure A-33 shows a few of the various diffuser shapes available.



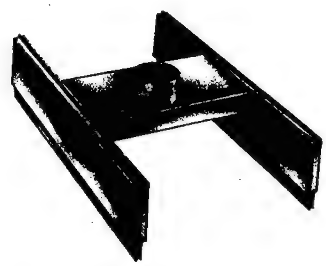
(a) Model TDC Square & Rectangular Ceiling Diffuser.



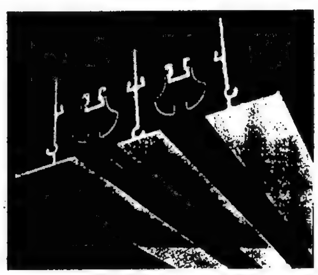
(b) Model TDC-NT Narrow Tee-Louvered Face Diffuser.



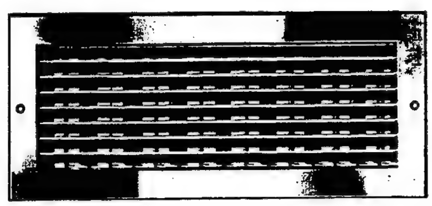
(c) Model PMC Perforated Ceiling Diffuser.



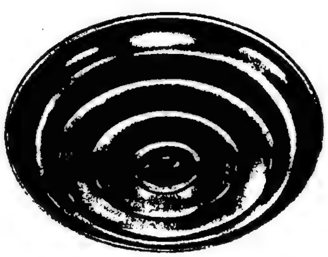
(d) Model LTT Fluorescent Light Troffer Diffuser.



(e) ML Series Linear Slot Aluminum Modulinear Diffuser.



(f) Model 301RL 300/350 Supply Grilles and Registers.



(g) Model TMRA Round Ceiling Diffuser.

Figure A-33. Diffuser Types.

Provided by Titus.

When the sound level is critical in selecting a diffuser, be aware that the noise generated by the diffuser will be substantially higher than the manufacturer's published data when a duct turn precedes the entrance to the diffuser or if a balancing damper is installed immediately before the diffuser.

The throw of a diffuser is the horizontal and vertical axial distances that the airstream travels at various velocities after leaving the diffuser. The throw values listed in a manufacturer's table are based on specified velocities. The room section in Figure A-34 illustrates the throw pattern of air leaving a ceiling diffuser. In this example, the selected diffuser with an airflow of 200 cfm has throw values of 15-19-27 based on terminal velocities of 150, 100, and 50 fpm, respectively. Several techniques are available for applying throw data. A few of these are illustrated in Figure A-35.

Figure A-36 is a typical manufacturer's chart, and illustrates the selection process for the diffuser shown in Figure A-33(e). In the example, the area under consideration requires (by design) 85 cfm per ft. The diffuser selected is a 4 slot linear diffuser with 100 cfm per foot, and throw values of 15-19-27. Using the room configuration from Figure A-35, and designing for heating and cooling, the throw values required are 7-11-19.

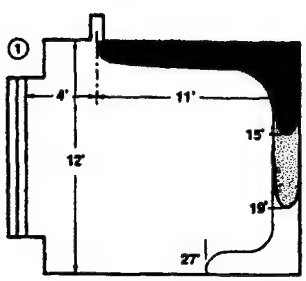


Figure A-34. Diffuser Throw.

Provided by Titus.

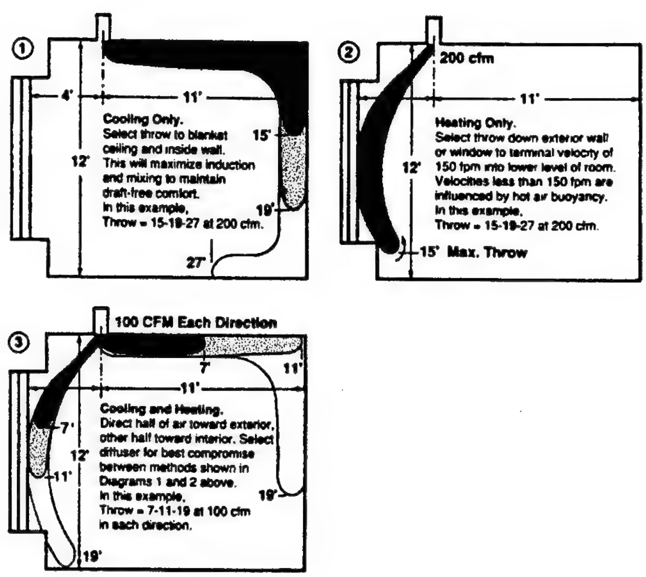


Figure A-35. Diffuser Throw Applications.

Provided by Titus.

Supply Performance Data for Plenum Applications

- All pressures are in inches of water.
- Throw values are based on an active section 3 feet long. For a 1 foot section the throw values are 0.6 times those shown. For a 10 foot or longer continuous length, the throw values are 1.6 times those shown.
- Horizontal (H) throw values are given for terminal velocities of 150, 100 and 50 fpm.
- Horizontal (H) throw values are based on air discharged in the same direction from all slots.
- For divided throw, use the cfm-per-foot value for the number of slots in each direction. For sound, use the NC values for cfm-per-foot for the total number of slots.
- Vertical (V) throw values are given for a terminal velocity of 50 fpm.
- NC values are based on a room absorption of 10 dB, re 10^{-12} watts, with a 10 foot active diffuser section (see the table of corrections below) and with horizontal throw. For vertical throw, deduct 11 dB from each NC value.
- For continuous lengths, the selection for NC is usually based on a 10 foot section.
- Dash (—) in space denotes NC value less than 10.
- Data were obtained from tests conducted in accordance with ISO Standard 5219, ISO Standard 3741 and ADC Test Code 1062 GRD84.

Model ML-37

No. of Slots	Total Press. H / V	.005 / .003	.019 / .012	.043 / 029	.075 / .050	.117 / .080	.170 / .114	.225 / .155	.290 / 200
1	CFM per Ft.	5	10	15	20	25	30	35	40
	Throw, Feet H	1-1-2	1-2-9	2-5-11	4-8-13	7-10-14	9-10-15	9-11-16	10-12-18
	Throw, Feet V	2	6	9	11	12	14	15	16
	NC	—	—	—	17	23	29	33	38
2	CFM per Ft.	10	20	30	40	50	60	70	80
	Throw, Feet H	1-1-4	2-4-12	5-8-15	8-12-17	10-14-19	12-15-21	13-16-23	14-17-24
	Throw, Feet V	4	8	13	16	17	19	20	22
	NC	—	—	13	22	28	34	38	43
3	CFM per Ft.	15	30	45	60	75	90	105	120
	Throw, Feet H	1-2-7	3-7-15	7-11-18	10-15-21	13-16-23	15-18-25	16-19-27	17-21-29
	Throw, Feet V	5	10	15	19	21	23	25	27
	NC	—	—	16	25	31	37	41	46
4	CFM per Ft.	20	40	60	80	100	120	140	160
	Throw, Feet H	1-2-9	4-8-17	9-13-21	12-17-24	15-19-27	17-21-29	19-23-32	20-24-34
	Throw, Feet V	7	12	18	22	25	27	29	31
	NC	—	—	18	27	33	39	43	48
5	CFM per Ft.	25	50	75	100	125	150	175	200
	Throw, Feet H	1-3-10	6-10-19	10-15-23	13-19-27	17-22-30	20-24-33	21-26-35	23-27-38
	Throw, Feet V	7	14	20	25	28	30	32	35
	NC	—	—	19	28	34	40	44	49
6	CFM per Ft.	30	60	90	120	150	180	210	240
	Throw, Feet H	2-4-11	7-11-21	11-16-26	15-21-29	18-24-33	21-26-36	23-28-39	24-30-42
	Throw, Feet V	7	16	22	27	30	33	35	38
	NC	—	—	20	29	35	41	45	50
7	CFM per Ft.	35	70	105	140	175	210	245	280
	Throw, Feet H	3-6-16	8-11-23	12-18-28	16-23-32	20-26-36	23-28-39	25-30-42	26-32-45
	Throw, Feet V	8	16	24	29	32	35	38	41
	NC	—	—	21	30	36	42	46	51
8	CFM per Ft.	40	80	120	160	200	240	280	320
	Throw, Feet H	3-6-13	8-12-24	13-18-29	17-25-34	22-27-38	24-30-42	26-32-45	28-36-48
	Throw, Feet V	8	17	26	31	35	38	41	43
	NC	—	10	22	31	37	43	47	52

NC Correction for Various Diffuser Lengths

Length, Feet	1	2	4	8	9	10	15	20	25	30
Supply	−16	−11	−6	−3	−2	0	+3	+5	+6	+8
Return	−10	−7	−4	−2	−1	0	+2	+3	+4	+5

Figure A-36. Supply Performance Data for Plenum Applications.

Provided by Titus.

Slot diffusers are often used for VAV systems. This type of diffuser distributes the air in a uniform pattern. The direction and volume of the discharge air can be adjusted.

Properly selected VAV diffusers, due to their design, will maintain proper air circulation at varying discharge flow with minimum effect on horizontal throw, and with minimum air dumping.

Dampers

A damper is a device used to vary the volume of air passing through an air outlet, air inlet, or duct. Tight-closing dampers provide energy savings by eliminating leakage of hot or cold air. Several manufacturers have designed dampers that are capable of shutting off a very high percent of air leakage. Some have accomplished this by having rubber seals on the ends of the damper blades, which, when fully closed, interlock and allow little air to pass between the blades. Air leakage may occur, however, around the outside of the blades between the frame and blades. Several types of dampers are available for HVAC systems. A few of them will be discussed in brief.

Motorized Control Dampers: Dampers that open or close to divert, direct, or shut off airflow in the primary duct system. Blades should have sealing edges using felt, rubber, etc. to ensure a tight cutoff of the airstream when closed.

Back-Draft Dampers: Back-draft dampers close under the action of gravitational force when no air is flowing and open when there is a drop in pressure across the damper in the direction of desired air flow. This prevents an undesirable backward flow of air or back-draft.

Multi-Shutter Damper: The parallel blade damper will deflect airstream when the damper is partially open. It is used to adjust air volume only when airstream deflection is acceptable.

Multi-Louver Round Diffuser Damper: This damper is a series of parallel blades and adjusts the air volume to the space.

Opposed Blade Round Diffuser Dampers: A series of pie-shaped blades are mounted in a round frame to adjust the air volume.

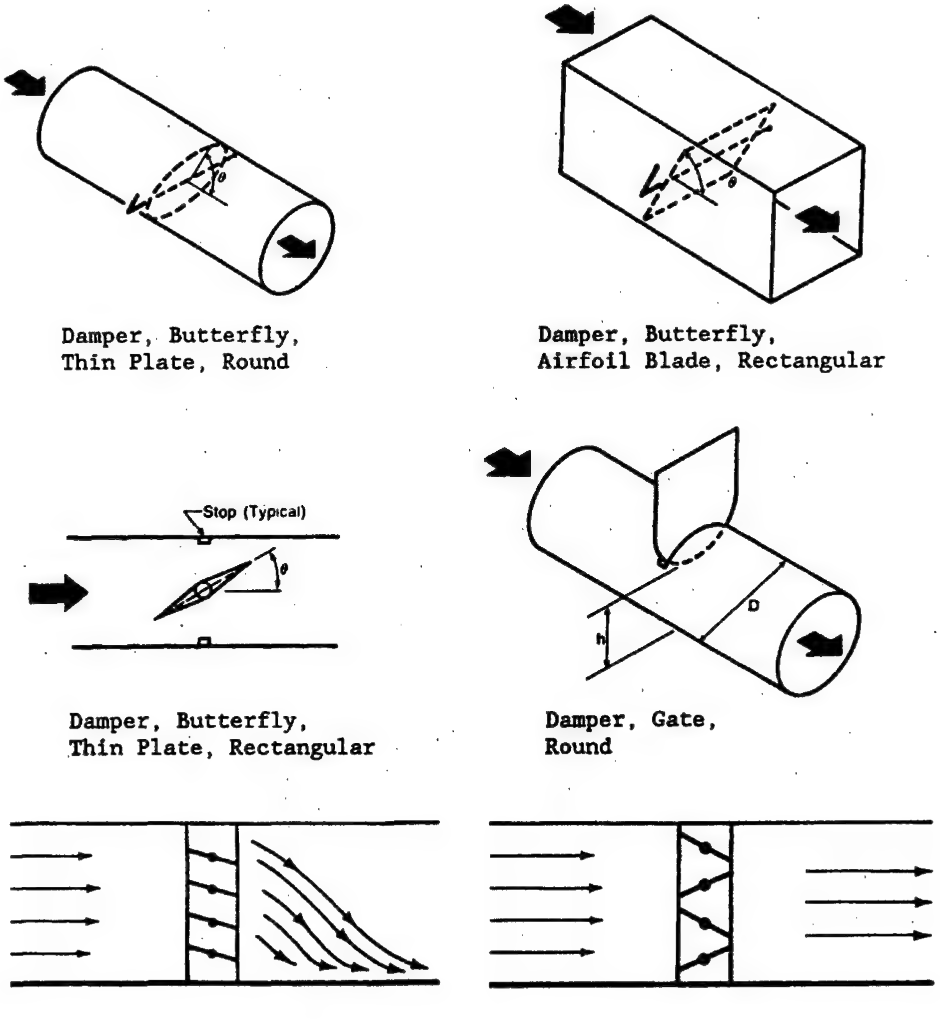
Diffuser Splitter Damper: This damper is a single plate hinged at the duct branch connection to the outlet and is used only with an equalizing device to adjust volume to the space.

Figure A-37 illustrates various types of dampers.

Volume control dampers come in "opposed" blades or "parallel" blades. When it is partially closed, the parallel blade damper diverts the air stream to the side of the duct (Figure A-37, bottom left). This will cause a non-uniform velocity profile

beyond the damper, and flow to close branch ducts on the downstream side may be seriously affected. Use of an opposed blade damper is recommended when volume control is required at the fan outlet, and there are other system components such as coils or branch takeoffs downstream of the fan. When the fan discharges into a large plenum or to a free space, a parallel blade damper may be satisfactory.

One important final point in considering dampers, obstructions in the duct work, and other components of the HVAC system (for purposes of calculating pressure drops, etc.) is to remember the placement of required fire dampers.



Damper, Butterfly, Thin Plate, Round

Damper, Butterfly, Airfoil Blade, Rectangular

Damper, Butterfly, Thin Plate, Rectangular

Damper, Gate, Round

Parallel Bladed Damper Illustrating Diverted Flow

Opposed Bladed Damper Illustrating Non-Diverted Flow

Figure A-37. Damper Types.

SMACNA, *HVAC Systems Duct Design*, 3rd Ed., 1990. Used with permission.

5 VAV Controls

A heating, ventilating, and air-conditioning (HVAC) system, its control system, and the building in which they are installed should be viewed as inseparable parts of a single entity. They interact in many ways, and neglecting any element may cause a partial or complete loss of controllability. It is important to consider that no HVAC system is better than its controls and the building in which it is installed.

In general, only three basic elements are necessary for a control system: sensor, controller, and controlled device. A sensor measures the temperature of the air and passes the information on to the controller. The controller compares the air temperature to a set point, then sends a signal to open or close the controlled device as required to maintain correspondence between the air temperature and the set point.

The HVAC system operates most economically when equipment capacity is closely matched to load. This may be accomplished better with an automatic control system than manually. A completely automatic system with changeover controls, interlocks, and internal monitoring and compensating controls minimizes human intervention and the chance of human error.

Control systems must have some source of energy to make them work. There are six sources generally used to cause control actions:

Electric Systems: Electric systems provide control by starting and stopping the flow of electricity, or varying the voltage and current by means of rheostat or bridge circuits.

Electronic Systems: These systems use very low voltages (24 V or less) and currents for sensing and transmission, with amplification by electronic circuits or servo-mechanisms as required for operation of controlled devices.

Pneumatic Systems: Pneumatic systems usually use low-pressure compressed air. Changes in output pressure from the controller will cause a corresponding position change at the controlled device.

Hydraulic Systems: These are similar in principle to pneumatic systems but use a liquid or gas rather than air. These systems are usually closed, while pneumatic systems are open (some air is wasted).

Fluidic Systems: Fluidic systems use air or gas and are similar in operating principles to electronic as well as pneumatic systems.

Self-Contained System: This type of system incorporates sensor, controller and controlled device in a single package. No external power or other connection is required. Energy needed to adjust the controlled device is provided by the reaction of the sensor with the controlled variable.

In actuality, what is controlled is the environment of enclosed spaces. As the environment's temperature, humidity, and pressure levels fluctuate, they become the "controlled variable." Control actions depend first on measurement of the controlled variable. Accurate and rapid measurement is the most serious problem in the control industry. It is difficult to get an accurate instantaneous reading, especially if the controlled variable is fluctuating or changing very rapidly.

The materials of which the sensors are made, their placement in the system, and the source of energy used for response may slow down the process that is so critical for accuracy. For example, thermostats will be affected by the presence or absence of air motion, temperature of the surface on which they are mounted, radiant heat from windows, etc. A pressure sensor placed at a point of turbulence (turn point or change in pipe size) in fluid cannot provide accurate or consistent readings.

Delay due to the distance over which the signal is transmitted is another problem that will arise in measurement or control. Pneumatic signals will travel only at sonic speeds and are subject to fluid friction losses. Electric signals may become seriously attenuated by resistance in long lines. These are only a few of the obvious problems encountered when designing a control system. Someone with a deeper knowledge and background in thermodynamics and electrical theory will more readily identify the problem areas in the design process.

The function of a military building may change several times in its existence and eventually will come the complaints of a bad thermal system design. Typically, when the environment is not properly controlled, the blame is placed on the control system, but in many cases, it is due to the HVAC system or building itself. The building must be designed to allow the degree of environmental control required. A warehouse cannot be used as a clean room, yet similar extremes have been experienced. This brings out the fact that a sophisticated control system will not compensate for the errors of a poorly designed HVAC system and will cost more in the future to maintain.

The following sections will discuss in greater detail the operation of thermostats, system control, and direct digital control.

Thermostats

Many instruments can be used for measuring temperature. Table A-8 lists some of these with a brief explanation of each.

Explanation of Thermostat Operation

In Figure A-38, a flapper-nozzle operation is shown. The thermostat provides a branch line airflow (pressure) that is a function of the ambient temperature in the room or controlled space. The force of the temperature sensing bimetal acting on the flapper is balanced by the feedback force of the pilot pressure acting on the opposite side of the flapper through the nozzle. When this force changes due to temperature or set point change, the position of the flapper changes over the nozzle and a new pilot chamber pressure is created. This pilot pressure feeds into the valve unit flow amplifier, which converts the low capacity pilot pressure to a higher capacity branch line change.

Table 1 Temperature Measurement

Measurement Means	Application	Approximate Range, °F	Uncertainty, °F	Limitations
Liquid-in-glass thermometers				
Mercury-in-glass	Temperature of gases and liquids by contact	−36/1000	0.05 to 3.6	In gases, accuracy affected by radiation
Organic	Temperature of gases and liquids by contact	−330/400	0.05 to 3.6	In gases, accuracy affected by radiation
Gas thermometer	Primary standard	−456/1200	Less than 0.02	Requires considerable skill to use
Resistance thermometers				
Platinum	Precision; remote readings; temperature of fluids or solids by contact	−430/1800	Less than 0.0002 to 0.2	High cost; accuracy affected by radiation in gases
Rhodium-iron	Transfer standard for cryogenic applications	−460/−400	0.0002 to 0.2	High cost
Nickel	Remote readings; temperature by contact	−420/400	0.02 to 2	Accuracy affected by radiation in gases
Germanium	Remote readings; temperature by contact	−460/−400	0.0002 to 0.2	
Thermistors	Remote readings; temperature by contact	Up to 400	0.0002 to 0.2	
Thermocouples				
Pt-Rh/Pt (type S)	Standard for thermocouples on IPTS-68, not on ITS-90	32/2650	0.2 to 5	High cost
Au/Pt	Highly accurate reference thermometer for laboratory applications	−60/1800	0.1 to 2	High cost
Types K and N	General testing of high temperature; remote rapid readings by direct contact	Up to 2300	0.2 to 18	Less accurate than listed above thermocouples
Iron/Constantan (type J)	Same as above	Up to 1400	0.2 to 10	Subject to oxidation
Copper/Constantan (type T)	Same as above, especially suited for low temperature	Up to 660	0.2 to 5	
Ni-Cr/Constantan (type E)	Same as above, especially suited for low temperature	Up to 1650	0.2 to 13	
Beckman thermometers (metastatic)	For differential temperature in same applications as in glass-stem thermometer	10°F scale, used 32 to 212°F	0.01	Must be set for temperature to be measured
Bimetallic thermometers	For approximate temperature	−4/1200	2, usually much more	Time lag; unsuitable for remote use
Pressure-bulb thermometers				
Gas-filled bulb	Remote testing	−100/1200	4	Caution must be exercised so that installation is correct
Vapor-filled bulb	Remote testing	−25/500	4	Caution must be exercised so that installation is correct
Liquid-filled bulb	Remote testing	−60/2100	4	Caution must be exercised so that installation is correct
Optical pyrometers	For intensity of narrow spectral band of high-temperature radiation (remote)	1500 and up	30	
Radiation pyrometers	For intensity of total high-temperature radiation (remote)	Any range		
Seger cones (fusion pyrometers)	Approximate temperature (within temperature source)	1200/3600	90	
Triple points, freezing/melting points, and boiling points of materials	Standards	All except extremely high temperatures	Extremely precise	For laboratory use only

Table A-8. Temperature Measurement Instruments.

Reprinted with permission from the 1997 *ASHRAE Handbook*.

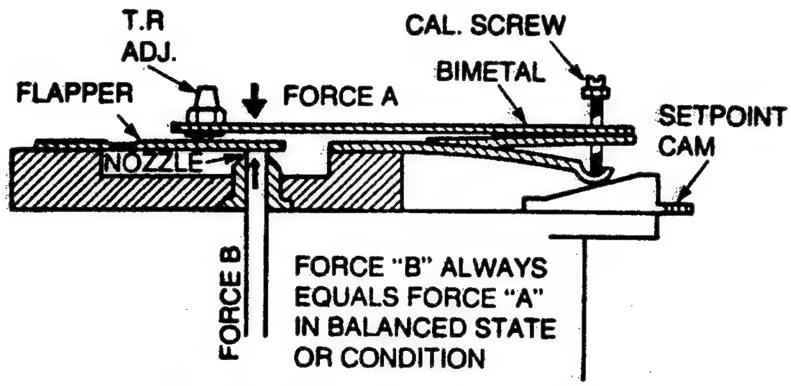


Figure A-38. Flapper-nozzle Operation.

Reproduced with the permission of the National Environmental Balancing Bureau, December 1996.

In this thermostat, a feedback feature at the nozzle provides a pressure regulating effect that negates the effect of normal air supply functions on the branch line.

Applications

Thermostats serve several purposes other than the single, simple-purpose type:

- The *day, night, or dual room* thermostat controls at a reduced temperature at night. Some electric types have an individual clock and switch built into the thermostat. A pneumatic "day-night" thermostat uses a two-pressure air supply system.* Changing the pressure at a central point from one value to the other actuates switching devices in the thermostat and indexes them from day to night or vice versa.

- *Heating-cooling, summer-winter* thermostats may have their action reversed, and if desired, may have their set point changed by the indexing means. Pneumatic "heating-cooling" thermostats use a two-pressure air supply.

- The *multi-stage* thermostat is arranged to operate two or more successive steps in sequence.

* Air pressure is varied through the same tube for summer and winter.

- A *submaster* thermostat has its set point raised or lowered over a predetermined range, in accordance with variations in output from a master controller.

- *Wet bulb* thermostats are used for humidity control with proper control of the dry bulb temperature. A wick or other means for keeping the bulb wet and rapid air motion to assure a true wet bulb measurement are essential.

- A *dewpoint* thermostat is a device designed to control from dewpoint temperatures.

- A *variable volume temperature system* monitor thermostat (made by Carrier only) communicates with as many as 36 other zone thermostats and has a digital display for set points. This thermostat uses the equation $q_s = 1.08 \times time \times cfm \times \Delta T$.

- *Discriminating* thermostats are electronic two-stage cooling, two-stage heating control devices. The discriminating thermostat uses solid state devices to turn on its stages and should be able to operate any air conditioning system that does not require special switching.

The location of these space controllers must represent the conditions of the whole zone served by the controller. In large open areas that have more than one zone, thermostats should be in the middle of their zone to assure they are not affected by conditions in the surrounding zones. Three different locations for space temperature controllers are in common use:

- *Wall-mounted thermostats* are usually placed on inside walls or columns in the occupied space they serve. Outside wall locations should be avoided. Thermostats should not be mounted where they will be affected by heat from other sources such as direct rays of the sun, pipes or ducts in the wall, convectors, direct air currents from diffusers, etc. The location should provide ample air circulation unimpeded by furniture or other obstructions, and should afford protection from mechanical injury. They should never be placed in spaces such as corridors, lobbies, foyers, etc., unless used only for control of these areas.

- *Return air thermostats* can be used for control of floor-mounted unitary conditioners such as induction fan-coil units and unit ventilators. On induction and fan-coil units, the sensing element will be located behind the return air

grille. On classroom unit ventilators that use up to 100 percent outdoor air for natural cooling, however, a forced flow sampling chamber will normally be provided for the sensing element. If return air sensing is to be used with central fan systems, the sensing element should be located as near the space being controlled as possible to eliminate influence from other spaces, and the effect of any heat gain or loss in the duct. Where combination supply/return air light fixtures are used to return air to a ceiling plenum, the return air opening of a light fixture can be used as a location for a return air sensing element. Precaution should be taken in location of the sensing element to avoid radiant effect, assure adequate air velocity across the element, and provide means for future access for servicing.

- *Diffuser-mounted thermostats* usually have sensing elements mounted on ceiling supply diffusers of the circular or square type and depend on aspiration of room air into the supply airstream. They should be used only on high aspiration diffusers that are adjusted for a horizontal air pattern. The diffuser in which the element is mounted should be in the center of the occupied area of the zone being controlled.

VAV System Control

Control system components, when combined to control an HVAC system, are defined as control loops. These control loops are further defined into two types, the open loop and closed loop.

An open loop control system usually takes corrective action to offset effects of external disturbance on the variable of interest in the system. The action is sometimes called the feed-forward control because it is anticipating the effect of an external variable on the system. This type of control does not provide complete control from the space temperature viewpoint.

The closed loop control system has a controller that measures the actual changes in the controlled variable, and actuates the controlled device to bring about an opposite change, which is again measured by the controller. The corrective action is a continuous process until the variable is brought to a desired value within the design limitations of the controller. This system of transmitting information about the results of an action or operation back to its origin is known as feedback and makes true automatic control possible.

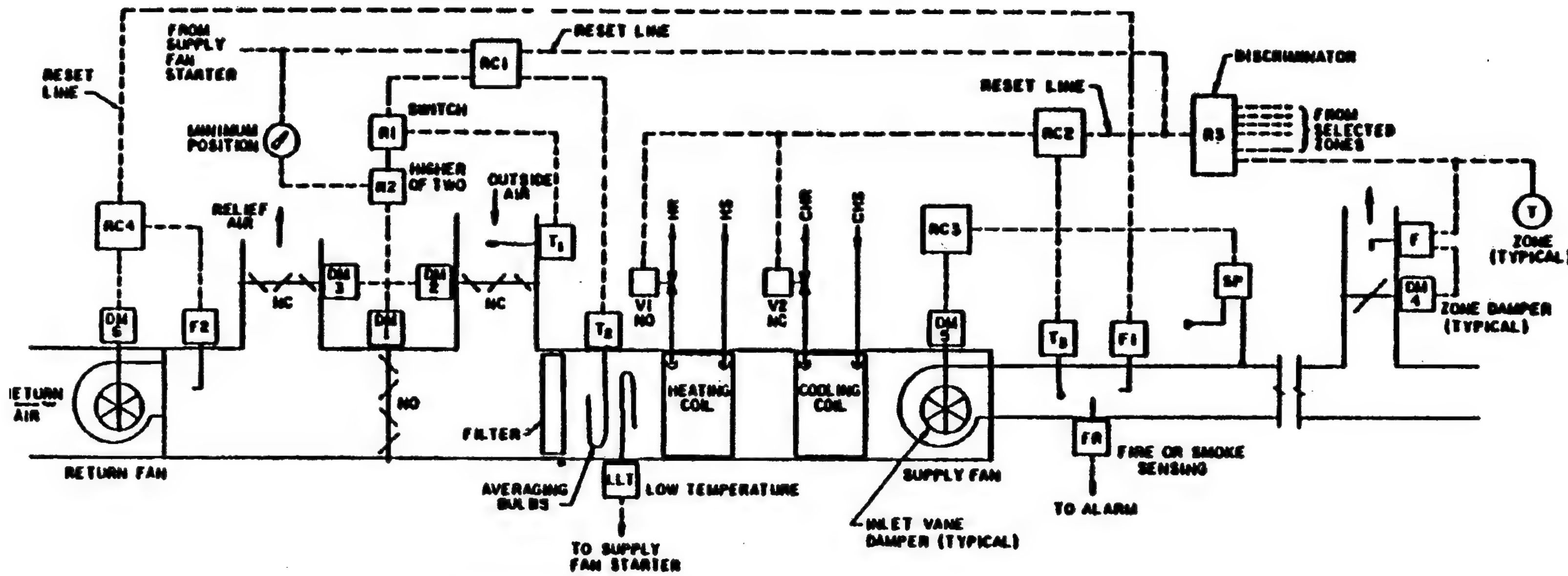
A subsystem is any system that is part of a larger system. An HVAC system may have several subsystems (i.e., a system to control pressure, one to control temperature, dampers, supply fan, etc). Each of these is part of the whole system. Having discussed the various components of a control loop and their function and explained what a control loop is, an example of controlling a VAV system will now be discussed. Figure A-39 shows the control diagram for a VAV system with an economy cycle and discriminator reset. This is perhaps as elaborate as is practical using conventional control hardware.

The control system has five essentially separate control subsystems, although they all interact. The first, on the right of Figure A-39, is the loop controlling the zone damper. Figure A-40 shows an expanded view of this control subsystem and shows a reheat coil that might be needed in exterior building zones. The "inner" or most direct loop here consists of the flow sensor, which modulates the zone damper to control the flow to the zone under varying duct pressures. Another "outer" loop consists of the room thermostat, which changes the set point of the flow control loop to vary the temperature. When reheat is required, the room thermostat has direct control over the reheat valve.

The next subsystem, to the left of the zone damper loop, is controlling the supply fan air volume. This is perhaps the simplest loop in the system. The differential pressure sensor, SP, transmits a signal to the receiver/controller RC3 which compares it to the receiver/controller set point. The output from RC3 controls whatever type of fan pressure control system is used (e.g. variable speed drive, inlet vanes or discharge dampers).

The next loop to the left is the loop that modulates the heating and cooling coil in sequence to control the fan discharge air temperature (T3 through RC2 to valves V1 and V2). Note that this "inner" loop has its set point changed by an outer loop consisting of the room thermostat in the hottest zone, which is adjusting the set point of the supply air temperature loop to control room temperature. Also note that valves V1 and V2 should operate in sequence to avoid wasteful simultaneous heating and cooling.

The economy cycle loop is essentially the same as the heating and cooling coil loop and also has a "reset" signal from the hottest zone. However, the two-position action of T1 acts through R1 to switch the damper position control loop in and out of service depending on outdoor air temperature.



LEGEND:

DM	Receiver controller	V	Valve
RC	Receiver controller	HR	Hot water return
F	Flow sensor	HS	Hot water supply
NO	Normally open	CHR	Chilled water return
NC	Normally closed	CHS	Chilled water supply
T	Thermostat	R	Discriminator reset

Figure A-39. Example VAV System Control Diagram.

Reprinted with permission from *Automatic Temperature Control for Energy / Cost Effectiveness*, ASHRAE Energy Professional Development Series.

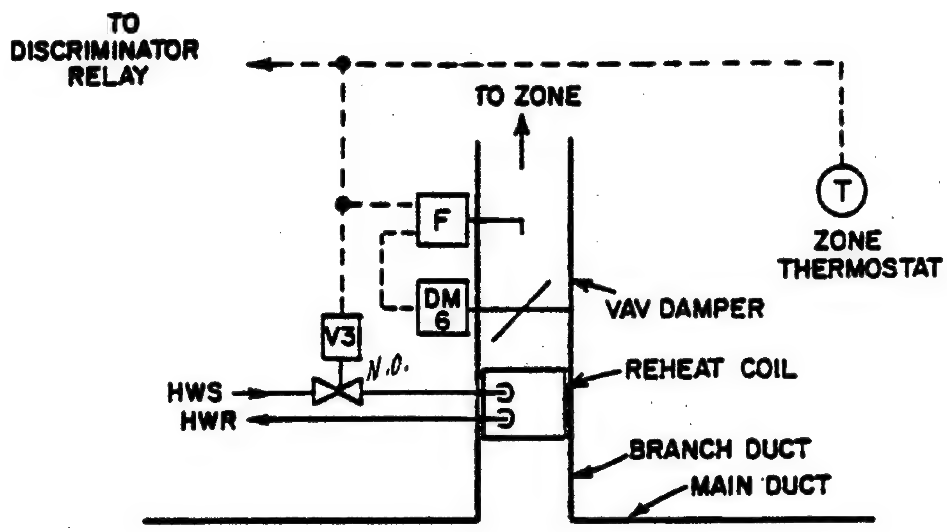


Figure A-40. Zone Damper Control Loop.

Reprinted with permission from *Automatic Temperature Control for Energy / Cost Effectiveness,* ASHRAE Energy Professional Development Series.

There is a return fan control loop where the signal from flow sensor F2 is compared with the "set point" in RC4, and the output of RC4 modulates the return fan. The "set point" is a reset signal from F1.

The last return fan control system is an example of some of the less-than-precise terminology (some of which is in quotes above) used in describing HVAC control systems. In this case, when the fan speed or damper position changes, flow sensor F1 reads essentially the same change in flow as F2, and it responds just as quickly. Neither sensor signal is really a "reset" signal in this case. Both represent equally important and similar feedback paths, and both feedback signals occur without delay.

Direct Digital Control (DDC)

The following overview is given to familiarize the reader with DDC systems so that they will not be intimidated by DDC if they encounter it in the field. If DDC is used, this information will also help provide an understanding of its applications and operations.

An analog computer is defined as a computer that operates with numbers represented by directly measurable quantities. Analog control systems are standard

today and are basically understood by most specifiers, installers, users, and maintenance personnel. The idea of changing to a new, digital system could be of great concern and an area of resistance for many personnel.

Many factors should be considered in selecting a DDC system: price, feature, service availability, track record of distributor, track record of manufacturer, and history of dependability. To avoid pitfalls, beware of unknowledgeable manufacturers, designers, specifiers, suppliers, installers, operators, and maintenance personnel. To prioritize these factors, one must determine the criticality of the process to which the DDC system is to be applied. In addition to the preceding considerations, one must beware of a system manufactured without assurance of direct replacement compatibility. Many products, especially electronic or computer products, can become obsolete before the normal building project can be completed. They are purposely designed to be discarded rather than repaired upon failure.

DDC is considered to be the automatic control or condition of a process by a digital computer. The size of the computer varies from a very small microprocessor to a central microcomputer. The controlled device (damper operator, valve operator, stepping relays, etc.) may be pneumatic, electric modulating, or two position. (It is often said that the system will be either pneumatic or DDC. This means that the system may be DDC and pneumatic, with the DDC system replacing the pneumatic controller but operating the same pneumatic-controlled device.)

The DDC system uses a combination of computer hardware components and software (computer disks, etc.) to maintain the controlled variable (temperature, pressure, relative humidity, flow rate, level, etc.) according to the desires of the operator. Instead of calibrating the hardware controller, the control sequence and set point are input to the computer by software and modified by a keyboard entry at the operator's console.

The DDC system monitors the controlled variable (temperature, pressure, etc.) and compares the value stored in the computer. When the measured value is not equal to the desired value, some systems output a series of digital pulses that are converted to the controlled device by an electric-to-pneumatic transducer. Transducers maintain the output signal until readjusted by the computer. Other systems may change the control signal by a series of flip flops (on-off or open-close signals) to bleed air into or out of a pilot relay or the control line. Some interfacing signal device will always be required to isolate the computer output circuitry from the control signal circuitry.

Keyboard functions and the printouts should be clearly defined, and the printouts and indications should be easily interpreted so the operator will be quickly advised of an incorrectly inserted command. Just because a system is labeled user friendly does not mean it will be at the level any operator can interpret. Therefore, prospective buyers should investigate before purchasing.

Some of the advantages of using a DDC system are listed below:

- *Energy Optimization:* A deadband or zero energy band can be programmed into the system, thereby reducing the energy consumption for heating and cooling. This requires no additional hardware because the system already requires control interconnection to the heating and cooling controlled devices. Other energy saving routines are:
  - Close outside air damper when outside air temperature exceeds a predetermined point
  - Close outside air damper during building warm-up, prior to occupancy time
  - Open outside air damper for pre-occupancy cool down
  - Close cooling and heating water valves prior to system shut down at the end of the day
  - Conservation of central plant energy by use of proper control strategies to optimize operation of chillers, boilers, heat exchangers, etc.
  - Precise control of controlled variable when desired
  - Heuristic control is available on most well-designed DDC systems. This feature is useable for warm up or cool down (optimum start).
- *Competitively Priced:* For today's computer products, the trend has been that prices go down.
- *Multi-Schedule Capabilities:* During a single day, there could be several set points (temperatures) for the building. Also, the system may be programmed to cycle the equipment off during lunch time.

- *Remotely Accessible:* With the aid of a telephone modem, many DDC systems may be accessed for readouts and adjustments from any remote location that has a telephone by the use of a computer input terminal keyboard or a separate properly programmed computer. This permits a single operator to monitor many systems and buildings from a single location.

- *Unauthorized Tampering Prevented.*

In the end, the final user holds the ultimate key to the successful operation of the total DDC system. If a perfect system is designed, a perfect product is chosen, a perfect installation is performed, and a perfect control strategy is programmed and commissioned, but the user does not care to understand or maintain the system in its perfect state, the whole system will fail. User education must start early to ensure satisfaction for all parties to the project.

6 Acoustical Considerations

In considering the acoustical performance of an HVAC system, it is important to develop a vocabulary of acoustical terminology. The following definitions will aid in understanding the proceeding overview of acoustical considerations for VAV systems.

- *Aerodynamic Noise:* Also called generated or self-generated noise, is noise of aerodynamic origin in a moving fluid arising from flow instabilities. In duct systems, aerodynamic noise is caused by airflow through elbows, dampers, branch wyes, pressure reduction devices, silencers, etc.

- *Airborne Noise:* Noise that gets to observer by transmission through air.

- *Background Noise:* The irreducible noise level measured in the absence of any building occupants.

- *Breakout Noise:* The transmission or radiation of noise from a part of the duct system to an occupied space in the building. Also referred to as "flanking" and "duct radiation."

- *Noise:* Sound which is unpleasant or unwanted by the recipient.

- *Room Effect:* The difference between the sound power level discharged by a duct (through a diffuser or other termination device) and the pressure level heard by an occupant of a room.

- *Sound Power Level (L_w or PWL):* The fundamental characteristic of an acoustic source (fan, etc.) is its ability to radiate power. Sound power level cannot be measured directly; it must be calculated from sound pressure level measurements. Sound pressure level is comparable to the measurement of temperature in a room, whereas the sound power level is comparable to the cooling capacity of the equipment conditioning the room. The resulting temperature is a function of the cooling capacity of the equipment, and the heat gains and losses of the room. In the same way, the resulting sound pressure

level would be a function of the sound power output of the equipment together with the sound reflective and sound absorptive properties of the room.

- *Structure Borne Noise:* This condition is apparent when sound waves are being carried by a portion of the building structure. This noise is translated from radiation of structure borne sound into the air.

Many sources of noise are found in VAV systems. Noises transmitted through the duct system are from fan, aerodynamics, and duct termination devices. Fan manufacturers usually provide sound power level data in octave bands. If this information is unavailable, accurate data may be calculated if volume flow rate, static pressure, and type of fan are known by use of equations and charts. These equations and charts are located in literature such as *ASHRAE Fundamentals* handbooks and SMACNA's *HVAC Systems Duct Design*.

Aerodynamic noise is generated when airflow in a duct system becomes turbulent as it passes through sharp bends, sudden enlargements or contractions, and other devices that cause a substantial pressure drop. Aerodynamic noise is important to consider at velocities above 2000 fpm in the main ducts, 1500 fpm in branch ducts, and 800 fpm in ducts serving room terminal devices. When duct system velocities are as listed above or when the duct does not follow good airflow design principles, aerodynamic noise may become a major problem. Aerodynamic noise is usually from 31.5 through 500 Hz octave band center frequencies, which are low frequencies. Because of the magnitude of low-frequency energy, it transmits readily with not much loss through light gauge walls of ducts, and through suspended acoustical ceilings.

With regard to duct terminal devices, pressure reducing valves in mixing and variable volume boxes usually have published noise ratings indicating the sound power levels that are discharged from the low pressure end of the box. Manufacturers may specify the requirements, if any, for sound attenuation materials between the outlet and box in the low pressure duct. If boxes are located away from critical areas such as mechanical rooms, janitorial closets, or corridors, the noise radiating from the box may be of no concern. If the box is located above a critical space, and separated from the space by a suspended acoustical ceiling that has little or no transmission loss at low frequencies, the radiated noise from the box may exceed the noise criterion for the room below. In such a case, relocate the box or encase it with a construction having a high transmission loss.

Room air devices such as diffusers, grills, light fixtures, and air handling ceiling suspension bars are always rated for noise generation. The room air terminal unit is selected to meet the noise criterion (N.C.) required or specified for the room, bearing in mind that the manufacturer's sound power rating is obtained using a uniform velocity distribution throughout the diffuser neck or grill collar. When balancing dampers are installed immediately before the diffuser, or if a duct turn precedes the entrance to the diffuser, airflow will be turbulent and the noise generated by the device will be substantially higher than the manufacturers published data.

A misalignment or offset that exceeds approximately one-quarter diameter in a diffuser collar length of two diameters can also cause a significant change in diffuser sound power level above that of the manufacturer's published data. Figure A-41 shows an example of increased pressure drop and increased noise level for a flexible duct connection. When there is an offset of only 1/8 the diameter, there is no appreciable change in the diffuser performance.

Most grills and diffusers are furnished with integral volume dampers. Since dampers generate noise when partially closed, the sound power levels of the units are a result of the air volume handled by the diffuser, and the magnitude of the pressure drop across the damper.

Another interesting acoustical consideration is the noise path between adjacent rooms caused by short lengths of duct. This is called "cross talk." The most common method of controlling cross talk is to avoid connecting rooms with short lengths of duct, by lining the ducts connecting these rooms with acoustical materials, and by installing silencers (sound absorbing devices) in the duct.

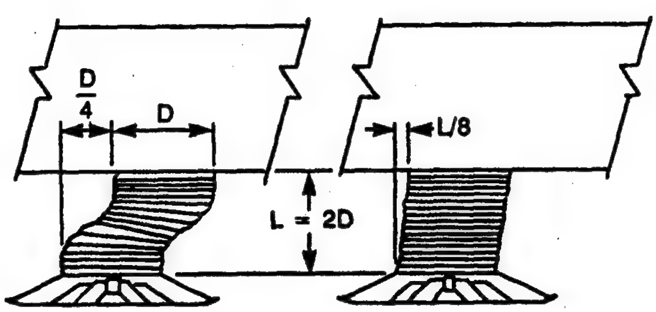


Figure A-41. Example of Increased Pressure Drop and Noise Level for Flexible Duct Connection.

Courtesy Dwyer Instruments, Inc.

Many of these acoustical considerations for VAV systems are the same for other HVAC systems. Those which are of a more critical nature for the VAV system have been discussed briefly here. Considerations for acoustical improvement on most HVAC systems are: (1) components such as fan and duct system vibration isolation, (2) more in depth study of breakout noise, duct system noise; and how to calculate each, and (3) materials and methods used to prevent acoustical problems, such as duct lagging, to prevent breakout noise. Many charts and publications from sources such as SMACNA are available for these calculations.

7 Acceptance Testing

Acceptance testing of a new facility's VAV system can be an important step in ensuring the system's correct and energy efficient operation. Although current construction practices should be capable of providing functional VAV systems for heating and cooling, there is no guarantee that this will happen. Corps quality assurance (QA) representative(s) have to rely, for the most part, on the competence of the contractor's quality control (CQC) to provide a proper system. Acceptance testing will assist the Corps in determining the quality of the contractor's testing, adjusting, and balancing (TAB) work. The acceptance testing procedures should be easy to use by field personnel but still provide accurate results.

Normally, after the HVAC system, including the VAV subcomponents, for a new facility has been installed by the contractor or his subcontractor, it will be tested, adjusted, and balanced by a TAB subcontractor to ensure proper operation. The TAB subcontractor documents his work in a TAB report, which includes the data collected during TAB, as well as the air flows, fluid flows, temperatures, etc. established for the system. Upon completion of TAB, the report is submitted to the Corps QA representative for review and approval.

CERL has already developed an acceptance test procedure for air supply and distribution systems. This procedure provides a relatively detailed series of acceptance tests for the HVAC system's fans, ducts, and coils, but does not cover the VAV boxes that may also be in the system. This deficiency will be corrected here.

To provide a better understanding of the TAB process, the following sections will first describe the instruments commonly used in TAB, followed by a brief description of basic TAB procedures. The final section will discuss acceptance testing of VAV systems.

Air Measuring Instruments

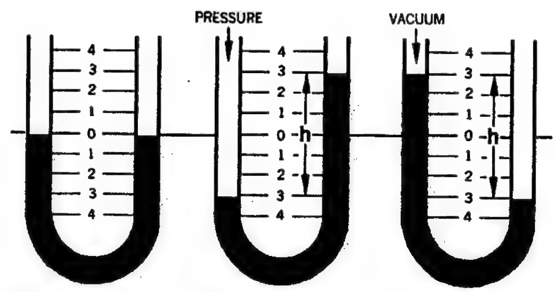
Many different types of testing and measuring instruments are used in TAB work. These instruments are used for measuring pressures, temperatures, fluid flows, electrical circuits, and rotational speeds. Many of these instruments are expensive, and must be protected from dirt as well as shock and jarring. Some of the instruments can be damaged by exceeding their rated capacity. The National Environmental Balancing Bureau (NEBB) has established criteria for calibration of those instruments that require it. The most common air measuring instruments are described below.

Manometers

U-Tube Manometer: This is the basic instrument for measuring air system pressures. It is a practical instrument to use, and is inherently 100 percent accurate. The U-Tube manometer consists of a U-shaped tube about half filled with liquid.

When a positive pressure is applied to one leg, as shown in the Figure A-42, it pushes the liquid down in one side and up in the other. The difference in the height (h) indicates the pressure.

When a vacuum is applied to one leg, the difference in height (h) indicates the amount of vacuum.



In its simplest form the manometer is a U-tube about half filled with liquid. With both ends of the tube open, the liquid is at the same height in each leg.

When positive pressure is applied to one leg, the liquid is forced down in that leg and up in the other. The difference in height, "h," which is the sum of the readings above and below zero, indicates the pressure.

When a vacuum is applied to one leg, the liquid rises in that leg and falls in the other. The difference in height, "h," which is the sum of the readings above and below zero, indicates the amount of vacuum.

Figure A-42. Measuring Pressure and Vacuum With a Manometer.

Courtesy Dwyer Instruments, Inc.

Electronic Meters

These manometers (Figure A-43) are battery powered and contain no fluid. They are accurate, small and light, come with a digital display, and are priced competitively with liquid-filled manometers.

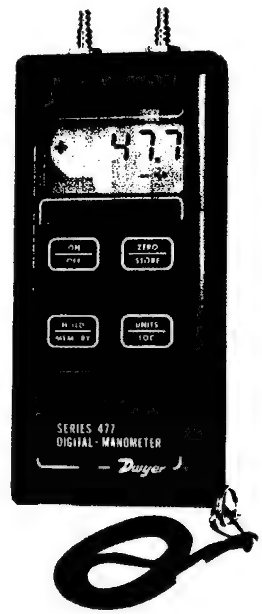


Figure A-43. Electronic Meter.
Courtesy Dwyer Instruments, Inc.

Dry-Type Pressure Gauges (Magnehelic)

These gauges (Figure A-44) are small, lightweight, inexpensive, and relatively easy to use. They contain a diaphragm that readily moves with changes in the pressure imposed on it. Its movement is restricted by the range spring, which is calibrated to bend a definite amount when the diaphragm is subjected to a given pressure. One of the main advantages of the magnehelic gauge is that it does not contain liquid that may be spilled or can be blown out by pressure beyond the range of the instrument.

Figure A-44. Magnehelic Gauge.
Courtesy Dwyer Instruments, Inc.

Pitot Tube

The primary use of the Pitot tube (Figure A-45) is to measure the velocity in ducts to determine the duct airflow (cfm). The pitot tube can be used to measure any one of the three basic pressures (total pressure, static pressure, and velocity pressure) when used with the proper hose hook-ups (Figure A-46).

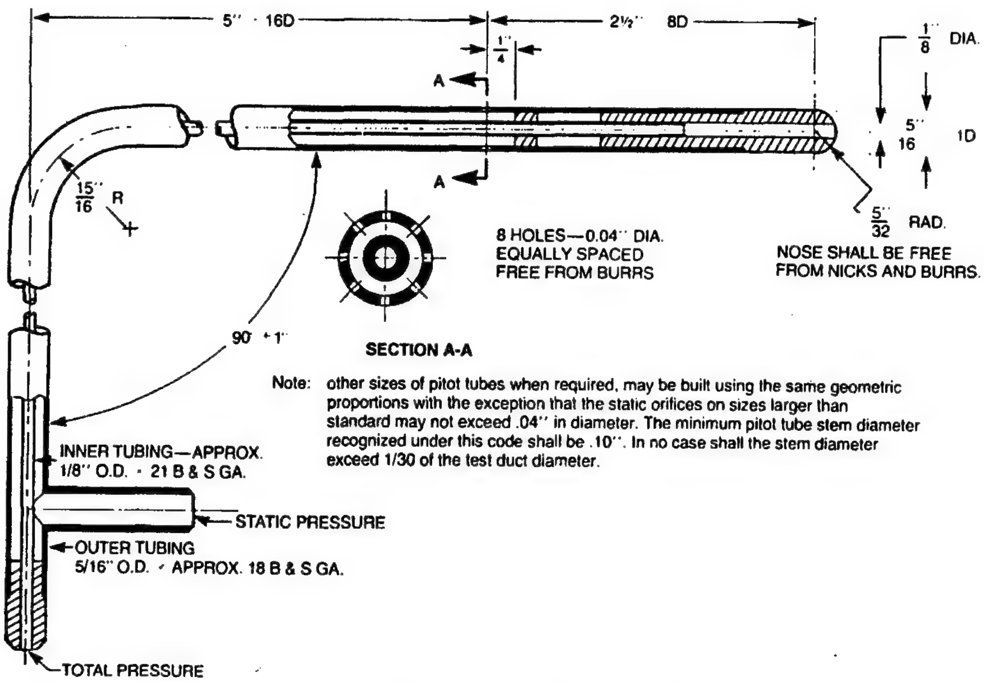


Figure A-45. Pitot Tube.

SMACNA, *HVAC Systems - Testing, Adjusting & Balance*, 2nd Ed., 1993. Used with permission.

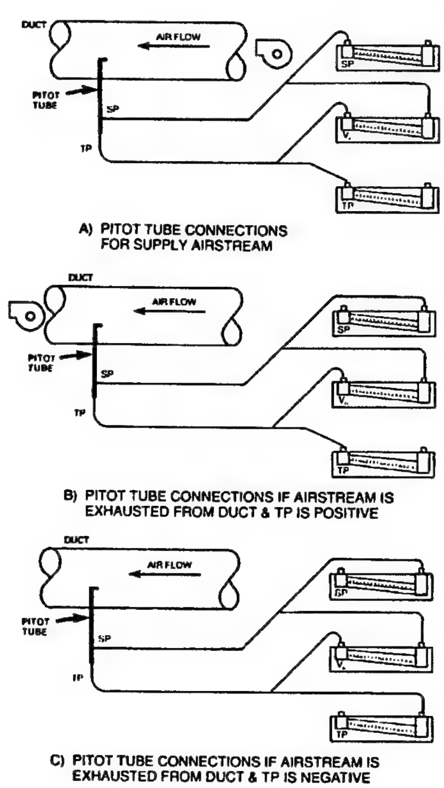


Figure A-46. Variation of Pitot Tube Connections.

SMACNA, *HVAC Systems - Testing, Adjusting & Balance*, 2nd Ed., 1993. Used with permission.

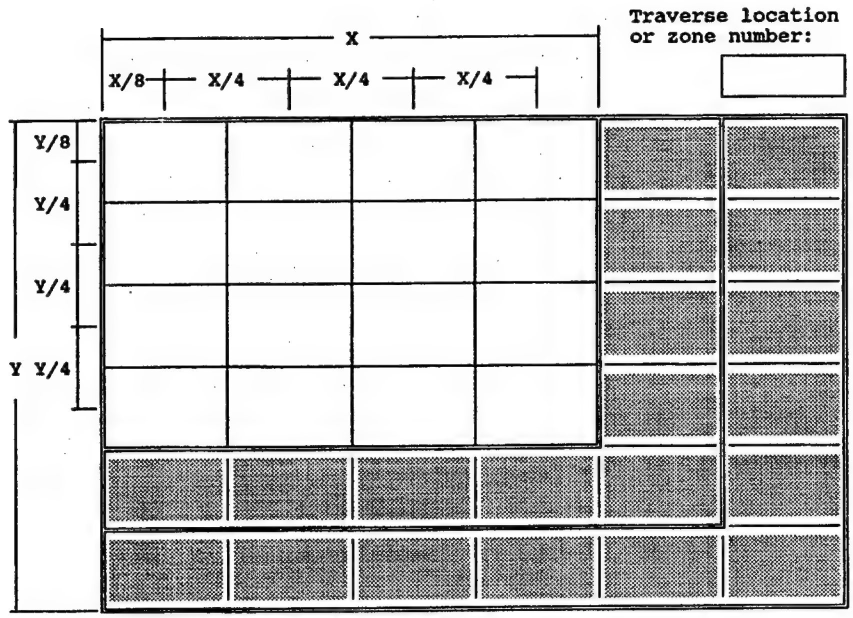
Use of the Pitot Tube: The correct procedure for measuring velocities in ducts are listed below:

Measure the size of the duct (free inside dimensions of the duct where the air is passing through), and determine the cross-sectional area (in sq ft).

To perform a pitot tube traverse of a duct, readings must be taken in the duct at equal intervals. Figure A-47 illustrates the procedure.

In round ducts, reading should be taken at the centers of equal concentric areas (Figure A-48). Preferably, 20 readings should be taken, with 10 along each of two diameters.

Use the worksheet on the next page to calculate volumetric air flow at fan outlet or zone. Divide duct to be measured into 16 blocks. Use manometer and pitot tube to take readings of velocity pressure at the centerpoint of each block. Convert the velocity pressure in each block to velocity using AABC or SMACNA conversion tables or the formula $V = 4005\sqrt{VP}$, where V is velocity in fpm, and VP is velocity pressure in inches of water.



Note: If the maximum distance between traverse points is greater than 6", expand the duct traverse diagram as necessary by using the shaded blocks.

$$\text{Average Velocity (fpm)} = \frac{\text{sum of readings}}{\text{number of readings}}$$

$$\text{Net Area (sq ft)} = \frac{\text{duct width} \times \text{duct height}}{144}$$

Volumetric Air Flow = Average Velocity × Net Area

	Average Velocity (fpm)	Net Area (sq ft)	Volumetric Air Flow (cfm)
Design			
TAB			
Actual			

Figure A-47. Duct Traverse Worksheet.

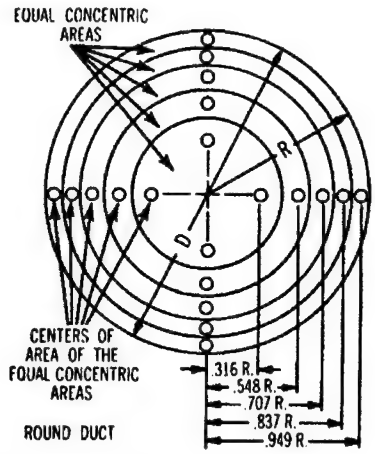


Figure A-48. Readings From Equal Concentric Areas of a Round Duct.

Courtesy Dwyer Instruments, Inc.

Rotating Vane Anemometer

The rotating vane anemometer (Figure A-49) consists of a propeller connected to a dial that is calibrated in feet. It is used for the measurement of supply, return, and exhaust air quantities at registers and grilles.

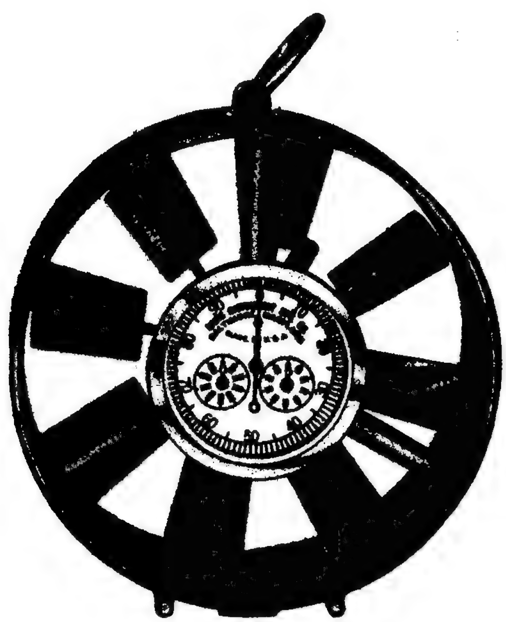


Figure A-49. Rotating Vane Anemometer.

Source: Davis Instrument Mfg. Co., Inc. Used with permission.

Hot-Wire Anemometer

This instrument (Figure A-50) is used to measure very low air velocities such as room air currents, and airflow in hoods and troffers. It can also be used for measurements at grilles and diffusers, although much less frequently than other velocity measuring instruments. It operates on the principle that the resistance in a wire will increase when heated. The probe is extremely directional and delicate and must be held at right angles to the airflow.

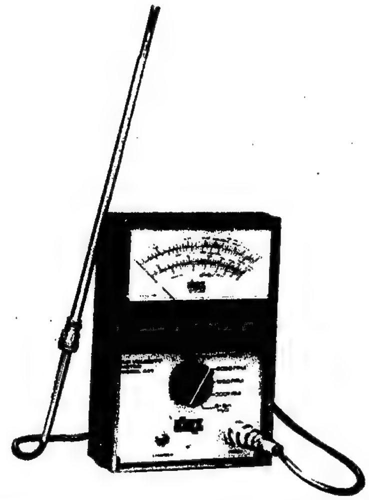


Figure A-50. Hot-Wire Anemometer.
Source: Davis Instrument Mfg. Co., Inc. Used with permission.

Flow Measuring Hood

The flow measuring hood (Figure A-51) is used to measure air distribution devices directly in cfm when balancing a large number of ceiling diffusers or balancing troffer diffusers. The flow hood is rapidly becoming the most popular instrument in the TAB industry for measuring airflow.

Figure A-51. Flow Measuring Hood.
Source: Shortridge Instruments, Inc. Used with permission.

Testing, Adjusting, and Balancing

Testing uses specialized instruments to measure temperatures, pressures, rotational speeds, electrical characteristics, velocities, and air and water quantities to evaluate the equipment and system performance.

Adjusting is the final setting of balancing devices such as dampers and valves. Adjusting also includes setting the automatic control devices such as thermostats and pressure controllers.

Balancing is the regulation of air flow through the system. The wrong amount of air flow through individual diffusers can lead to complaints of drafts and/or excessive noise. It can also cause stuffiness and poor temperature control.

Preparation for TAB should not wait until after construction has started. TAB should be considered **during design** of the HVAC system (see Chapter 3, p A-61). Only a trained technician from a certified TAB firm (normally AABC or NEBB) should carry out the TAB process. Certification requirements are generally provided in the construction specifications. The following information is given to provide an understanding of the TAB process.

Preliminary Steps

These steps are standard TAB practice and are the minimum that should be performed by the TAB firm. Failure to follow these may result in a poor TAB process. Corps QA representatives should confirm that these steps are actually completed.

1. Procurement of data:

   a. Contract drawings: Up-to-date contract drawings.

   b. Specifications: These will spell out exactly what data and testing are required, and what guidelines or balancing procedures are desired.

   c. Submittal data: Obtain all applicable approved equipment submittals, especially for the following equipment:

      i. Fans: Include performance data, physical characteristics, fan curves. Pay close attention to the external and total static pressure ratings of fans.

ii. Air terminals: Include air pressure loss at design flow conditions, sound pressure data, air pattern adjustment, and recommended testing procedure. Note whether the terminal has the means for airflow adjustment or if auxiliary dampers must be used.

iii. Air Distribution devices: List components and related capacities. List air distribution devices such as variable volume boxes and static pressure dampers.

iv. Primary heat exchange equipment: Performance data for boilers, chillers, cooling towers, and heat exchangers should be examined to ensure that unit capacity and pressure losses are within acceptable tolerances.

v. Terminal heat exchange equipment: Performance data for HVAC unit coils, reheat coils, fan coil units, and unit heaters should be reviewed to ensure temperature and pressure ratings comply with design requirements.

2. Review and Analysis of Systems:

   a. Review the plans, specifications, and equipment data.

   b. Prepare a schematic layout of each duct and piping system.

   c. Prepare the test report forms (normally AABC or NEBB).

   d. Complete systems checklist.

   e. Duct system checks. Observe the ductwork to see if it is complete and installed correctly. Does the installation match the plans? Is the system really ready for balancing? Verify that all terminal devices are installed and that their dampers are open. Inspect the system for leakage.

Final Testing, Adjusting, and Balancing Procedures

As the TAB subcontractors technicians go through their TAB work, they will generally follow the steps listed below. The Corps QA representative must be present during TAB to observe that all of these steps have been completed and performed correctly.

1. Assemble the previously prepared paper work, drawings, etc. for the system that you are starting on. Make sure that all preliminary procedures have

been performed. Check dampers and drives and verify that nobody is working in, on, or around the equipment to prevent injuries upon start up.

2. Using standard forms, set up AHU to design condition. Check:

   a. rotation

   b. rpm

   c. belts

   d. dampers

   e. motor voltage

   f. motor amps to verify no overload

3. For pressure independent systems, start air balance by cycling VAV box closest to fan through maximum and minimum air flows, filling out proper forms.

   a. Set at full open and measure airflow.

   b. Set at minimum open and measure airflow.

Continue until all boxes are cycled.

4. If system has no diversity, set all boxes at full cool.

   a. Read static pressure at AHU and fill out proper forms.

   b. Traverse supply duct to verify total airflow.

5. If system has diversity, set a required number of VAV boxes, as determined by reviewing the building heat gain/heat loss calculations, and determine the peak at maximum load time. Once the peak load boxes are selected, they are set for maximum flow, and the remaining boxes are set for minimum flow. This procedure simulates a maximum cooling situation within a building or a specific area of a building.

   a. Read static pressure at AHU and fill out proper forms.

   b. Traverse supply duct to verify total airflow.

6. If a system does not perform to specifications, as determined through the above balance procedures, some items which might indicate improper performance are:

   a. Low airflow

   b. High airflow

   c. Too high static pressure

   d. Too low static pressure

   e. Excess amps to motors

   f. Insufficient amps to motors

   g. Air noise.

The engineer and contractor should be contacted to perform necessary modifications to bring system up to design specifications.

NOTES:

A. If VAV system has pressure dependent VAV boxes, balancing becomes more complex since box airflow somewhat depends on duct static pressure. The general procedure is followed, except additional sequential readings normally are required to ensure proper airflow.

B. Diversity is when a single VAV system serves many areas within a building that do not reach their peak cooling load at the same time, thereby reducing the maximum airflow required at a specific time.

Acceptance Testing for Performance Verification

The Corps QA representative should perform the following checks or request that the TAB subcontractor perform them in the QA representative's presence. These checks will allow the representative to make a quick verification that the system has been set up and is operating as specified and designed.

1. General

   a. Vary room thermostat to check VAV box performance

b. Check accuracy of several room thermostats

c. Confirm supply air temperature in room

d. Check diversity of system

e. Confirm flow at several diffusers at minimum and maximum flow

f. Confirm building pressure is correct

2. Air handling unit, check:

   a. Rotational direction of motor and fan

   b. Belt tightness

   c. Motor name plate data

3. After setting system to operate at a maximum condition (set thermostat), check:

   a. Motor voltage

   b. Motor amps

   c. Rpm of motor and fan

   d. External static pressure

4. Verify performance of selected VAV box:

   a. Check VAV box inlet static pressure at several boxes. Compare with manufacturer's requirements.

   b. Set boxes to minimum flow and confirm system response

   c. Set selected VAV boxes to maximum flow by adjusting thermostat

      i. Measure pressure drop across VAV box (Figure A-52)

      ii. Determine cfm using chart on side of VAV box

   d. Confirm proper VAV box damper operations at several locations

The data checked above should be compared against the measurements recorded by the TAB subcontractor in the TAB report. The worksheet provided at the end of this chapter may be used for this purpose. Significant discrepancies or deviations between these measurements and those in the TAB report indicate a potential problem in installation or adjustment of the system. The causes for the discrepancies or deviations must be determined, and corrected by the TAB subcontractor.

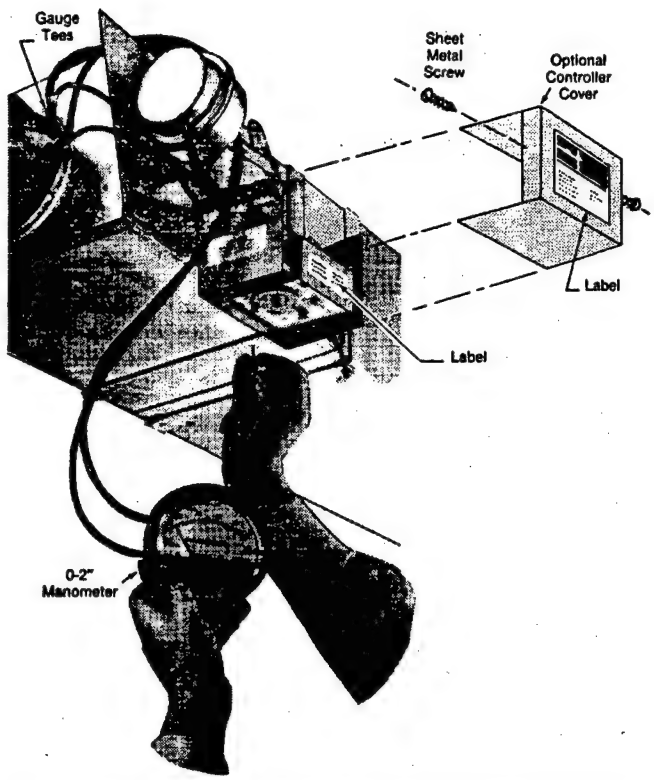


Figure A-52. Measuring Pressure Drop Across VAV Box.

Provided by Titus.

VAV SYSTEM ACCEPTANCE TESTING CHECKLIST

PROJECT:____

LOCATION:_______________________________________

NAME:____

General	Correct yes	Correct no	Date Checked
1. Vary room thermostats to check VAV box performance			
2. Check accuracy of several room thermostats			
3. Confirm supply air temperature in room			
4. Check diversity of system			
5. Confirm flow at several diffusers at min. and max. flow			
6. Confirm building pressure is correct			

Air Handling Unit Checks	Correct yes	Correct no	Date Checked
1. Rotational direction of motor			
2. Rotational direction of fan			
3. Belt tightness			
4. Motor name plate data (verify)			

Air Handling Unit Measurements	Design	TAB	Actual
1. Rpm of motor			
2. Rpm of fan			
3. Motor voltage			
4. Motor amps			

Air Handling Unit External Static Pressure	Design	TAB	Actual
1. Inlet side of AHU mixing box			
2. Outlet of AHU			
3. Traverse of discharge duct			

VAV Box Measurements

Box No.	Measurements	Design	TAB	Actual
	1. VAV box inlet static pressure			
	2. Pressure drop across VAV box			
	3. cfm			
	1. VAV box inlet static pressure			
	2. Pressure drop across VAV box			
	3. cfm			
	1. VAV box inlet static pressure			
	2. Pressure drop across VAV box			
	3. cfm			

Appendix A Annex: Humidification

The Need to Humidify

Control of relative humidity (RH) is essential for maintaining comfortable working conditions, proper functioning of sophisticated machinery and office equipment, and most important, the efficient use of energy.

If the temperature of air is increased without adding moisture to the air, RH is decreased. When RH is allowed to decrease significantly, either as a result of a heating system or from heat generated as a by-product of working machinery, lighting, or other energy released, a “dry air” condition is created. This dry air literally sucks the moisture out of everything—people, places, and things. For instance, dry air causes a migration of moisture from hygroscopic materials such as wood, paper, textiles, leather, or food. When these materials lose their moisture to dry air, physical changes of shrinkage, cracking, and hardening occur.

Dry air causes problems that range in severity from merely annoying, to extremely expensive, to dangerously life-threatening. In people, dry air sucks the moisture out of their respiratory systems, making them susceptible to colds, sore throats and other respiratory problems. It causes dry, flaky skin problems, and it generates static electricity in their hair (making hair unmanageable), in their clothes (making them uncomfortable), and their bodies (causing shocks). In controlled environments such as computer rooms, research laboratories, and industrial "clean rooms," static electricity can create serious problems. Static electricity can ruin computer programs that are stored on electronically sensitive tapes and disks. It also attracts dirt and dust in clean rooms where sensitive materials and products are manufactured. Hospitals must maintain high levels of humidity to control static electricity in the presence of high levels of oxygen and other potentially explosive gases.

An important consideration in military installations is the level of productivity. Dry air feels cold, even at higher temperatures. When people feel cold and uncomfortable, they are demonstrably less productive. Consequently, building temperatures (and energy costs) have to be raised in order for people to function

efficiently. But when the humidity level is correct, building temperatures can be lowered without changing the comfort level.

Psychrometrics

Air humidification and dehumidification are defined as the addition and subtraction of moisture from the air respectively. Each of these conditions is a change of state from liquid to gas or gas to liquid. Each occurs at a constant dry-bulb temperature, but of varying wet-bulb temperature. The same process is used for adding or subtracting latent heat. It also uses the same vertical line on Figure A1-1 at a constant dry-bulb temperature.

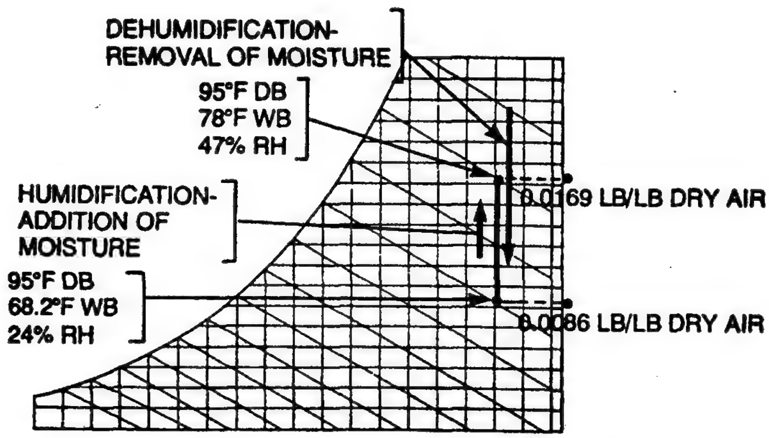


Figure A1-1. Humidification and Dehumidification.

Environmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

Humidification and dehumidification are both latent heat processes, and both are shown on the same chart.

In the following example, the only constant value is the dry-bulb temperature; all other properties increase for humidification and decrease for dehumidification. This process is an illustration and normally cannot be reproduced in environmental systems.

Example 1:

Find the amount of moisture required to increase the humidity of 1 lb of air at 95 °F and 24 percent RH to 95 °F and 47 percent RH.

Solution:

Plot the points on a psychrometric chart (Figure A1-1) and read the values on the right hand side:

95°F DB, 47% RH	=	0.0169 lb/lb dry air or (118 gr/lb)
95°F DB, 24% RH	=	0.0086 lb/lb dry air or (60 gr/lb)
Added moisture	=	0.0083 lb/lb dry air or (58 gr/lb)

Absorption

Absorption of steam into the air is very important. Some steam humidifier dispersion tube designs fail to mix the steam with the air adequately. Poor mixing of air and steam will result in slow absorption of steam into the air. This means trouble because unabsorbed steam collects on fans, dampers, turning vanes, and other obstructions in the duct. The results can be wet, leaking ducts, wet floors and equipment, and even algae and bacteria growth that could eventually be dispersed into the space being conditioned.

Steam goes through two changes of state as it becomes absorbed. When using steam injection or dispersion tubes in a duct, the absorption process is as follows. As the steam emerges from the dispersion tubes' discharge holes, it is invisible. The first change of state occurs as the steam almost instantly changes into a white fog of tiny water droplets. As the fog moves downstream and expands, it goes through the second change of state as it disappears. The second change of state should occur as quickly as possible, before anything in the duct gets wet.

The steam is turned into fog because the comparatively cool air that receives the steam becomes "supersaturated," causing excess moisture to condense and form a visible white fog. As it condenses, 970 Btu's of heat per pound of steam is released into the mixture of steam and air. As the visible fog moves downstream and fans out, it gradually re-absorbs this heat and re-evaporates, changing back into invisible vapor.

Fog Control

The air travel distance required for the two changes of state to occur is called the absorption distance or "fog travel distance." When using the dispersion tube method for injecting steam into the system, it is important to consider the distance at which these state changes take place. The conventional dispersion tube puts off steam, which requires a longer distance than a steam-to-steam dispersion tube. The conventional tube has a row of holes punched in the top of the tube. Condensate usually forms at these holes and flows down both inside and outside the tube. The outside condensate drips to the floor of the duct, and causes wetness problems.

In the steam-to-steam tube design, nylon inserts reach almost to the center of the dispersion tube so only the hottest, "driest" steam is discharged into the air stream. Any condensate on the inner walls of the tube collects on the tube bottom and drains back to the humidifier.

Figure A1-2 illustrates the air travel distance for conventional dispersion tubes vs. steam to steam dispersion tubes. Figure A1-3 shows sections of the steam-to-steam and conventional dispersion tubes, respectively.

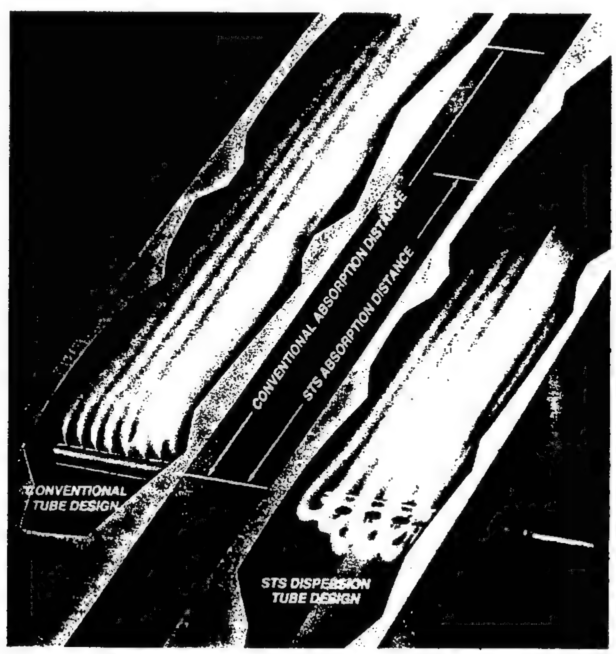
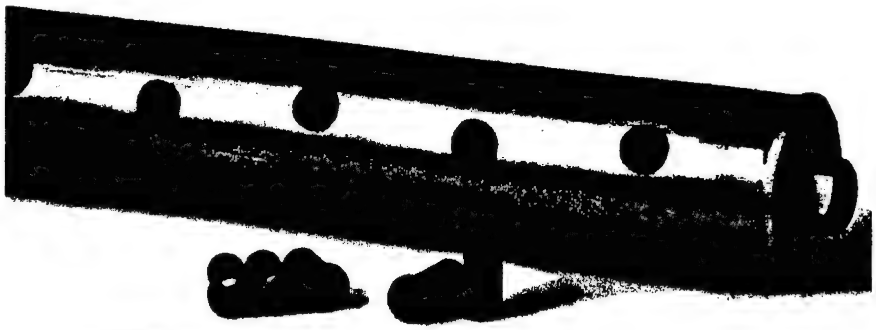
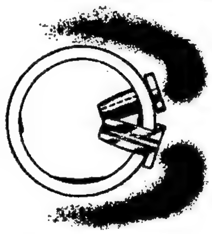


Figure A1-2. Air Travel Distance Comparison.





In the STS tube design, nylon inserts reach almost to the center of the dispersion tube, so only the hottest, 'driest' steam is discharged into the air stream. Any condensate on the inner walls of the tube is simply collected on the tube bottom and drains back to the humidifier.

A conventional dispersion tube has a row of holes punched in the top of the tube. Condensate usually forms at these holes and flows down both inside and outside the tube. Outside condensate drips to the floor of duct and causes wetness problems.

Figure A1-3. Sections of STS and Conventional Dispersion Tubes.

Humidifier Types

Area Humidifier (for wide open spaces)

This type of humidifier is used in spaces such as warehouses and shops. They operate as follows:

1. Steam enters separator from steam supply.
2. Condensate is removed and flows to steam trap in lower section of separator.
3. Dry steam rises through deflector plate into upper portion of separator surrounding inner re-evaporation chamber with steam at line pressure.
4. When humidifier valve opens, steam flows through piping into valve, through inner chamber, and through wood silencer into humidified space (any condensate passing through valve and piping is re-evaporated by line pressure steam jacketed inner chambers).
5. Electric fan disperses steam in room.

As fog is carried away from the humidifier by the fan air stream, it tends to rise toward the ceiling. If this fog contacts any solid surface (columns, beams, ceilings, pipes, etc.) before it disappears, it will probably collect and drip as water.

Figure A1-4 shows the various components of the area type humidifier.

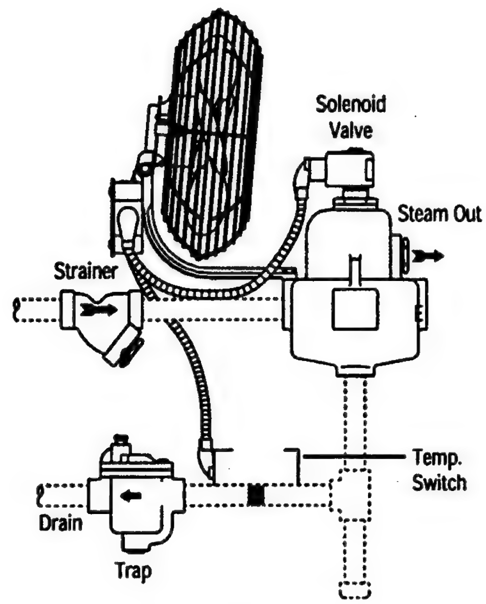


Figure A1-4. Components of an Area Humidifier.
Source: *Armstrong Humidification Handbook*, 1995. Used with permission.

Duct Type Steam Humidifiers

As mentioned before, area type humidifiers are used mainly in wide open areas. This application is not wise in places such as hospitals and small spaces with low ceilings. For these spaces, especially where VAV systems are used, duct type humidifiers are best. Generally, two types of injection humidifiers are used: single dispersion tube and multiple dispersion tube. These steam injection humidifiers are available to fit any size application. In most applications, the single dispersion tube humidifier is sufficient. It is commonly used when requirements for final duct relative humidity is below 60 percent or when there are 10 to 20 ft of straight duct downstream with no internal obstructions.

When conditions call for extremely short absorption distances, it is best to use a multiple dispersion tube unit. Two types of multiple dispersion tube units are

available for application. These are the Mini-Bank and Maxi-Bank humidifiers.*
For ducts of small cross-section, the Mini-Bank is used. The Maxi-Bank is for larger ducts and may be assembled on site.

Single dispersion tube humidifiers. In Figure A1-5, supply steam (1) circulates through the stainless steel jacket of the dispersion manifold (2) where its heat prevents condensation of steam being dispersed. From the manifold, supply steam flows into the separator chamber (3) through the directing nozzle (4) onto the conical separating baffle (5). Condensate is separated and flows to the bottom of the separator, and is discharged through the steam trap (6). Dry steam flows upward in the separator chamber, and is regulated by the control valve (7). Upon leaving the control valve, the steam passes through a final separator (8), which removes any condensate that may form, particularly on start up, and directs it back into the re-evaporation chamber (9) within the primary separator. Dry steam is then dispersed into the duct through the orifices of the steam jacketed stainless steel dispersing tube (10). A fine screen (11) covers the orifice to minimize noise. Figure A1-6 illustrates various mountings for single manifold humidifiers.

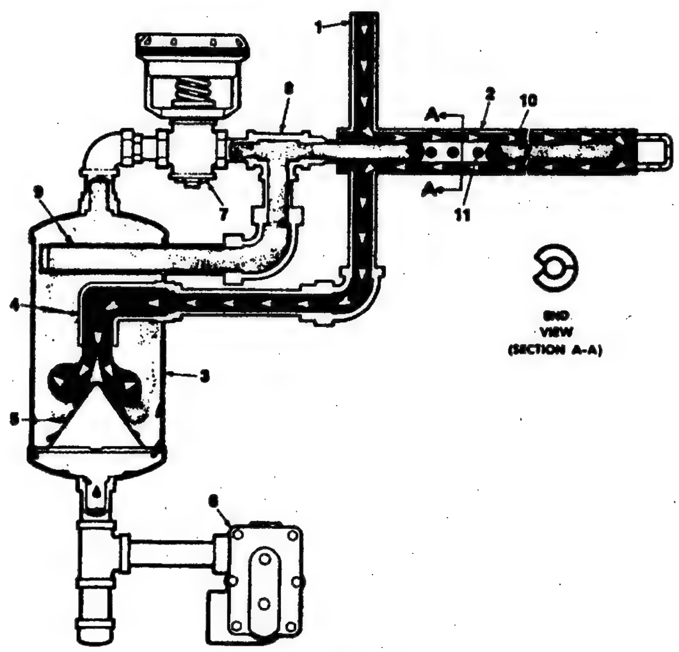
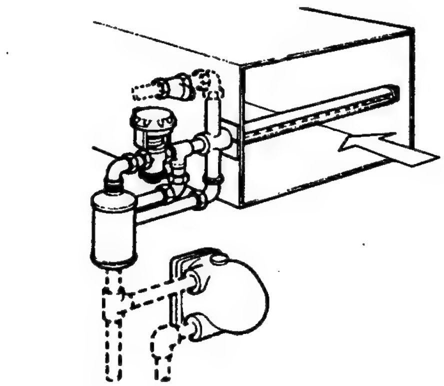
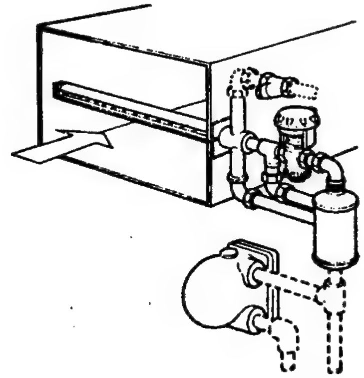


Figure A1-5. Single Dispersion Tube Humidifier.

* A commercial product of the "DRI-STEEM" Humidifier Company, Hopkins, MN.

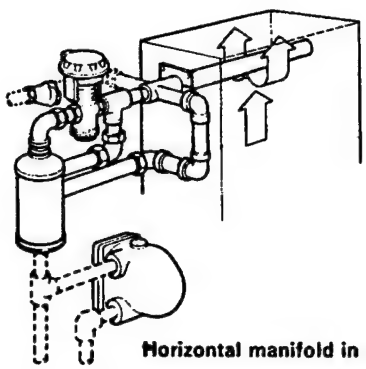


Typical Horizontal Installation
LEFT HAND
MOUNTED

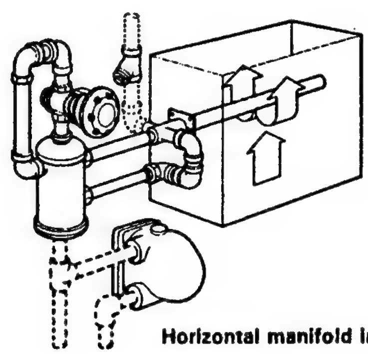


Typical Horizontal Installation
RIGHT HAND
MOUNTED

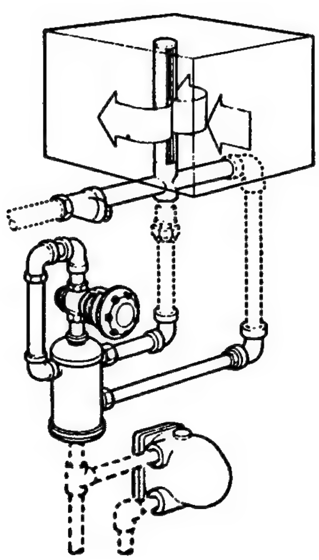
Optional Humidifier Hookups



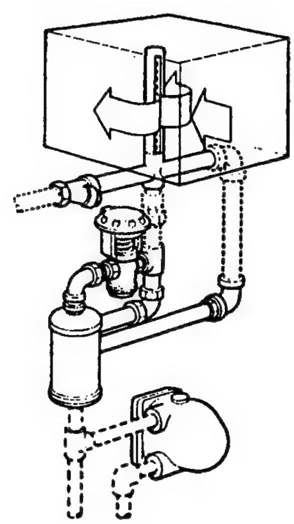
Horizontal manifold in vertical duct



Horizontal manifold in vertical duct



Vertical manifold installation



Vertical manifold installation

Figure A1-6. Various Mountings for Single Manifold Humidifiers.

Multiple dispersion tube humidifiers. A Mini-Bank multiple-tube humidifier is for small ducts up to 24 in. high and 48 in. wide. Because they are small, they are pre-assembled and ready for hook up.

The Maxi-Bank multiple-tube humidifiers are used when nearby downstream devices such as fans, coils, sound traps, and dampers could collect moisture and cause wetness problems in the air handling system. Formulas and tables for figuring the absorption distances with given temperatures and cfm are generally given with example problems for each model in the manufacturer's catalogs.

Trouble Shooting

Too Much Humidity

1. Humidity controller out of calibration.
2. Valve stem sticking.
3. Valve spring broken.
4. Foreign matter preventing valve from closing.

Too Little Humidity

1. Strainer screen plugged.
2. Stop valve not fully open.
3. Silencer media (steel wool) dirty.
4. Humidity controller out of calibration.
5. Inadequate steam pressure.
6. Undersized humidifier.
7. Automatic control valve not opening fully.
   a. Pneumatic valve operator leaking air.
   b. Valve stem sticking.

Humidifier Discharges Water

1. Faulty drainage:
   a. Return line pressure greater than humidifier pressure.
   b. Return line flooded.
   c. Dirty steam trap.
   d. Too much vertical lift.
   e. Wrong type steam trap; float type must be used.
2. Faulty steam supply:
   a. Humidifier supply not taken from top of main.

b. Too low (below 2 psi steam pressure).
c. Long, untrapped supply line.
d. Steam main flooded due to priming boiler.

Humidity Swings Above and Below Control Point

1. Boiler pressure swings too widely.
2. Faulty or inaccurate humidity controller.
3. Humidifier oversized.
4. Humidity controller in poor sensing location.
5. Pressure reducing valve not controlling accurately.

Glossary

AIR TERMINAL: See VAV BOX.

AIR VALVE ACTUATOR: A device which converts the control system signal into a useful function such as opening and closing the air valve.

ASPECT RATIO (DUCT): The ratio of length to width of an opening or core of a grille.

BEADING: The triangular or semi-circular crease in a duct (with spacing as indicated on page 3-24) that runs along the depth of the duct (not the length).

BLOCK TIGHT: Zero airflow into and out of the block.

BREAKOUT NOISE: Sometimes referred to as "flanking" or duct radiation. The transmission or radiation of noise from some part of the duct system to an occupied space in the building.

CLEAN ROOMS: A room designed to be as close to 100 percent free of contaminants as possible—hospital rooms, research laboratories, etc.

COMPRESSED AIR JET PRINCIPLE: As compressed air is forced through a nozzle or chamber of some kind, the air expands and provides a driving force.

CONDENSATE: The liquid formed by condensation of a vapor: in steam heating, water condensed from steam; in air conditioning, water extracted from air as by condensation on the cooling coil of a refrigeration machine.

CROSS BREAKING: Two creases (along the surface of the duct) that cross each other at the center of a rectangle, whose diagonals are the creases.

DAMPER: A device used to vary the volume of air passing through an air outlet, inlet, or duct.

DEADBAND: In HVAC, a temperature range in which neither heating nor cooling is turned on. In load management, a kilowatt range in which loads are neither shed nor restored.

DUCT AIR VELOCITY: High (2,500 fpm and up), Medium (1,100-2,500 fpm), Low (1,100 fpm and below).

DUCT LAGGING: A 1- to 2-in. thickness of glass fiber or rock wool covered with an air-impervious layer of material having a mass of at least 1 lb per sq ft. Its purpose is to dampen the duct surfaces and prevent contact between the duct surfaces and the mass layer.

DUCT RUNOUTS: The end of the ductwork, as in diffuser locations where air leaves the duct.

DUMPING DIFFUSERS: Dumping occurs when the volume of air leaving the diffuser is far less than the diffuser was designed for with respect to its location in a space. As a result, the cold air falls almost straight down, dumping its air just below the diffuser, and not accomplishing a mix with room air.

EXTENDED PLENUM: This is a trunk duct of constant size (usually at the discharge of a fan, fan coil unit, mixing box, constant volume box, etc.) extended as a plenum to serve multiple and/or branch ducts (see Semi-Extended Plenum).

EXTRACTOR: A device located at the tee of two ducts that is used to direct airflow. It is used mainly to divert air to branch takeoffs.

FAN BRAKE HORSEPOWER: The actual horsepower required to drive the fan. Included friction losses are due to air turbulence, inefficiencies in the fan, and bearing losses.

FAN DISCHARGE PRESSURE: The pressure read on a metering device when held right at the discharge face of the fan.

FRICTION LOSSES: Frictional losses are due to fluid viscosity, and are a result of momentum exchange between molecules in laminar flow and particles moving at different velocities in turbulent flow. Frictional losses occur along the entire duct length.

HEURISTIC CONTROL: A computerized control that proceeds along empirical lines, utilizing equations within the computer program.

HUMIDIFIER: A device to add moisture to air.

HUMIDIFYING EFFECT: The latent heat of vaporization of water at the average evaporating temperature times the weight of water evaporated per unit of time.

HUMIDISTAT: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

HUMIDITY: Water vapor within a given space.

HUMIDITY, ABSOLUTE: The weight of water vapor per unit volume.

HUMIDITY, PERCENTAGE: The ratio of the specific humidity of humid air to that of saturated air at the same temperature and pressure, usually expressed as a percentage (degree of saturation; saturation ratio).

HUMIDITY RATIO: The ratio of the mass of the water vapor to the mass of dry air contained in the sample.

HUMIDITY, RELATIVE: The ratio of the mol fraction of water vapor present in the air to the mol fraction of water vapor present in saturated air at the same temperature and barometric pressure. It approximately equals the ratio of the partial pressure of density of the water vapor in the air to the saturation pressure or density, respectively, of water vapor at the same temperature.

HUMIDITY SENSING ELEMENTS, ELECTRICAL: These cause a change in characteristics (resistance or capacitance) due to the hygroscopic nature of the elements used.

HUMIDITY SENSING ELEMENTS, HYGROSCOPIC: Change in size or form to cause a mechanical deflection. Some of the organic materials used are hair, wood, paper, or animal membrane, and man-made products such as nylon.

HUMIDITY, SPECIFIC: Weight of water vapor (steam) associated with 1 lb weight of dry air; also called humidity ratio.

INFILTRATION: The natural exchange of outside air directly into the building.

INTERFACING: A point or means of interacting between two systems.

INTERNAL HEAT GAIN: Heat generated from within the space being considered from sources such as people, lights, motors, etc.

JOURNAL BEARINGS: Bearings that support the shaft of a pump in the inlet or adjacent to it.

LATENT HEAT: The amount of heat necessary to change the quantity of water to water vapor without changing either temperature or pressure. Latent heat is removed when water vapor is condensed.

LEAVING AIR TEMPERATURE: Temperature of air after it leaves the cooling coil.

LOAD CONDITIONS: The actual physical state of the air within the space being controlled.

LOW PRESSURE (Air System): A forced air system that produces static pressures from zero to 2 in. of water column of pressure in a duct system.

MAGNEHELIC GAUGE: A pressure gauge used for air system work, and available in many different pressure ranges.

MASTIC: An aromatic resinous exudate from mastic trees; any of various pasty materials used as protective coatings or cement.

MICROCOMPUTER: A very small computer containing a microprocessor along with supporting devices such as a memory system.

MIXING BOX (Dual Duct): An inlet valve is positioned by a motor in response to a room thermostat to supply air at the proper temperature that satisfies the load within the space. This box is provided with warm air and cool air which are mixed in this box, and leave at a single temperature.

PEAK INSTANTANEOUS DEMAND: The greatest demand for a load at a given instant of time.

PILOT RELAY: A relay used for switching loads such as another relay or solenoid valve coils. The pilot relay contacts are located in a second control circuit. Rated in volt-amperes (VA).

PLENUM: An air compartment connected to one or more distributing ducts.

PRESSURE DEPENDENT SYSTEM: When no controls that compensate for changes in duct static pressure are utilized by the system, the air volume delivered is dependent on upstream static pressure changes.

PRESSURE INDEPENDENT SYSTEM: This system will deliver the required amount of air to satisfy the space load regardless of the changes in system static pressure.

SEMI-EXTENDED PLENUM: The semi-extended plenum is a trunk design system utilizing the concept of the extended plenum to incorporate a minimum amount of size reductions due to decreasing volume.

SENSIBLE HEAT: Heat that changes the temperature of the air without a change in moisture content.

SOLAR LOAD: The sun's rays contact the outer surfaces of a building (including window glass) generating heat on surfaces. This heat is transmitted through the walls and glazing at rates proportional to the insulating factors of each material. The transmitted heat becomes the solar load in the controlled space.

SPACE PEAK DEMAND: The greatest amount of kilowatts needed for a space during a demand interval.

SUPER HEATING: The difference between the temperature of a pure condensable fluid above saturation, and the temperature at the dry saturated state at the same pressure.

SURGE: The varying of airflow and static pressure due to an unstable condition in the duct system.

SYSTEM DIVERSITY: The ability of a single HVAC system to handle different (diverse) loads throughout a building that are all present at the same time.

THERMOMETER: An instrument for measuring temperature.

THERMOSTAT: An automatic control device actuated by temperature, and designed to be actuated by temperature.

THERMOSTAT, DIRECT ACTING: An instrument for activating a control circuit on sensing predetermined low temperature.

THERMOSTAT, REVERSE ACTING: An instrument for activating a control circuit on sensing predetermined high temperature.

THERMOSTATICALLY CONTROLLED: The use of a thermostat to send a signal to an operator which, in turn, will control a device or mechanism.

THROTTLED BACK: To reduce the flow of the medium being controlled by the mechanism (throttle).

TRUNK DUCT: The main duct from which branch ducts extend.

VARIABLE SPEED DRIVE: A device that varies the speed of a motor to match the load being put on the motor.

VAV BOX: Sometimes referred to as a VAV terminal. This box has controlled dampers inside that vary the volume of air sent to the controlled space.

VENTILATION: Bringing in outside air by use of a mechanical system.

ZONE: The specific section of a building controlled by a single thermostat. Buildings may be divided into many zones.

Abbreviations

AABC:	Associated Air Balance Council
AHU:	air handling unit
ASHRAE:	American Society of Heating, Refrigerating, and Air Conditioning Engineers
cfm:	cubic feet per minute
DDC:	direct digital control
DX:	direct expansion
fpm:	feet per minute
LAT:	leaving air temperature
MBtu:	mega British thermal units
MVR:	mechanical volume regulator
N.C. or NC:	normally closed
NEBB:	National Environmental Balancing Bureau
N.O. or NO:	normally open
OBP:	bypass operator
OMC:	modulating control operator
PDT:	pressure dependent terminal
PIT:	pressure independent terminal
RA:	return air
rpm:	rotations per minute
SA:	supply air
SMACNA:	Sheet Metal and Air Conditioning Contractors National Association
TAB:	testing, adjusting, and balancing
UPC:	Uniform Plumbing Code
VAV:	variable air volume
w.g.:	water gauge

Bibliography

Air Movement and Control Association, Inc. (AMCA), Publication 201-90 Fans and Systems, AMCA, Arlington Heights, IL, 1990.

American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), "Automatic Temperature Controls for Energy/Cost Effectiveness," ASHRAE Energy Professional Development Series, ASHRAE, Atlanta, GA.

ASHRAE, *Applications Handbook,* 1995.

ASHRAE, *Fundamentals Handbook*, 1997.

ASHRAE, *Systems and Equipment Handbook*, 1996.

Armstrong International, Inc., *The Armstrong Humidification Handbook*, Three Rivers, MI, 1995.

Carnes Company, Inc., Catalog, Verona, WI.

"DRI STEEM" Humidifier Company, *Humidifier Handbook*, Hopkins, MN.

Eppelheimer, Donald, "Coil Circuiting: A Key to Problem-Free DX/VAV Systems," *Contracting Business*, March 1988.

Guralnik, David B., editor in chief, *Webster's New World Dictionary, 2nd edition*, Prentice Hall Press, New York, 1986.

Honeywell, Inc., "Direct Digital Control System," Catalog of Controls for Heating, Ventilating, and Air Conditioning Systems, Wichita, KS.

National Air Filtration Association (NAFA), *NAFA Guide to Air Filtration,* NAFA, Washington, DC, 1993.

National Electrical Contractors Association, "Saving with Adjustable Speed Drives," Division 16/Electrical, Motor Control, National Electrical Contractors Association (NECA).

National Environmental Balancing Bureau, *Environmental Systems Technology*, National Environmental Balancing Bureau (NEBB), Vienna, VA, 1984.

NEBB, *Testing, Adjusting, Balancing Manual for Technicians*, April 1986.

Omnizone, "Modulating Zone Control Systems for VAV Applications of Air Conditioning Units," Huntington Beach, CA, May 1988.

PACE, "Central Station," Brod & McLung, Portland, Oregon.

Sheet Metal and Air Conditioning Contractors National Association, Inc. (SMACNA), *HVAC Duct Construction Standards--Metal & Flexible, 2nd Ed.* (SMACNA, Vienna, VA, 1995).

SMACNA, *HVAC Systems—Duct Design, 3rd Ed.*, 1990.

Spirax Sarco, "Steam Humidifiers," Bulletin 970, Allentown, PA, April 1985.

TempMaster - Air Distribution Products, "WATTMASTER" WCCII, Energy Management Service.

Titus Installation Manual, "Series ESV-3000 VAV Terminal," Division of Philips Industries, Inc., Richardson, TX.

Titus Products Catalog, "Diffusers and Terminal Units."

Trane Air Conditioning, *Varitrane Variable Air Volume Systems Manual*, The Trane Company, La Crosse, WI, 1978.

Appendix B: Boiler Systems

Principles, Applications, and Acceptance Testing

by
Dahtzen Chu
Energy Branch, Facilities Division
U.S. Army Construction Engineering Research Laboratory

and

Charles L. Burton
Alison J. Pacheco
Leland V. Speirs
Department of Architectural Engineering and Construction Science
Kansas State University

Contents

1 Introduction

The boiler is a closed vessel used to generate steam and hot water for heat or power. Within this vessel, water is contained and steam is produced and collected or hot water is produced. Heat is needed to change the water to the required medium. The most commonly used fuel sources for producing large volumes of steam or hot water are fuel oil, coal, or gas. Boilers come in many types and varieties. The following sections will describe in brief these various types of boilers, and the components they are composed of.

Boiler Types

Steam Boilers

A high-pressure steam boiler operates at pressures greater than 15 psig. One advantage of the high-pressure boiler is the reduced size of the boiler and steam piping. A low-pressure steam boiler operates at pressures less than 15 psig. An advantage of the low-pressure boiler is the simpler design and operation; no pressure reducing valves are required, and the water chemical treatment is less costly and complex.

Hot Water Boilers

A high-temperature hot water (HTHW) boiler furnishes water at a temperature greater than 250 °F or at a pressure higher than 160 psig. HTHW systems can carry greater heat to end locations than the lower temperature systems. A low-temperature hot water boiler furnishes water at a temperature less than 250 °F and a pressure less than 160 psig.

Hot water boilers usually require pumps to circulate the hot water and require power for pumping. Steam boilers do not require the pumps, but they do need larger piping. High-pressure steam systems will also require pressure reducing valves.

The efficiency of a boiler increases as the heating surface of the boiler increases. Figures B-1 and B-2 show that, with a larger heating surface, more heat is transferred to the water, and the amount of steam produced increases while using the same amount of fuel.

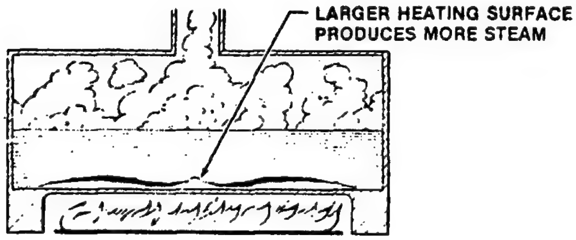


Figure B-1

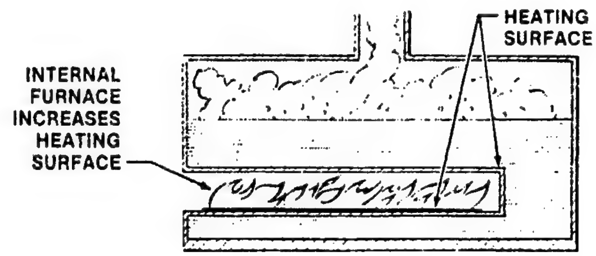


Figure B-2

Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

Boilers are classified based on their physical arrangement of the working fluid, the combustion gases, and the type of working fluid or heat carrier used.

Firetube Boilers

The largest percentage of small to medium-sized industrial boilers are firetube boilers (Figure B-3). The name comes from the tubes through which the flue gases flow. As the flue gases flow through the tubes, heat from the flue gases transfers to the water surrounding the tubes. Steam or hot water is generated in the process.

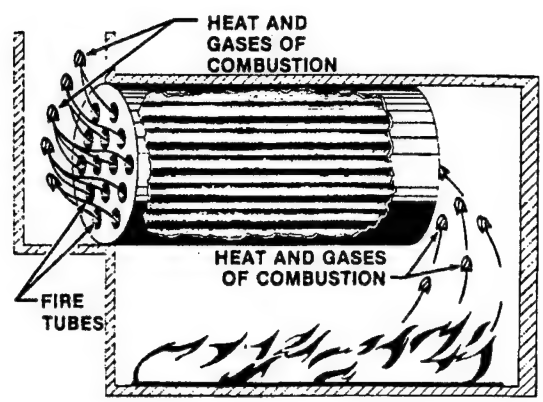


Figure B-3. Firetube Boiler.

Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

The most common firetube boilers used today are the Wetback and Dryback boilers. Both are variations of the Scotch boiler. Their names refer to the design of the rear of the combustion chamber, which is water-lined (Wetback) or lined with a high-temperature insulating material (Dryback).

The Wetback boiler (Figure B-4) has more heating surface, but is more difficult to service because of limited access.

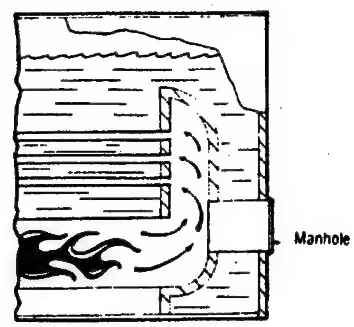


Figure B-4. Wetback Boiler.

Source: Dukelow 1983. Used with permission of Kansas State University, Manhattan.

The Dryback boiler (Figure B-5) is easier to service, but its insulation may deteriorate over a period of time, and its efficiency may be reduced if the insulation is not properly maintained.

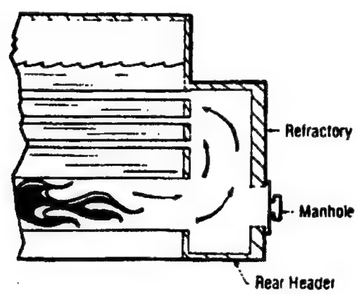


Figure B-5. Dryback Boiler.

Source: Dukelow 1983. Used with permission of Kansas State University, Manhattan.

The number of boiler passes for a firetube boiler refers to the number of horizontal runs the flue gases take between the furnace and the flue gas outlet. The combustion chamber or furnace is considered the first pass; each separate set of firetubes provides additional passes as shown in Figure B-6.

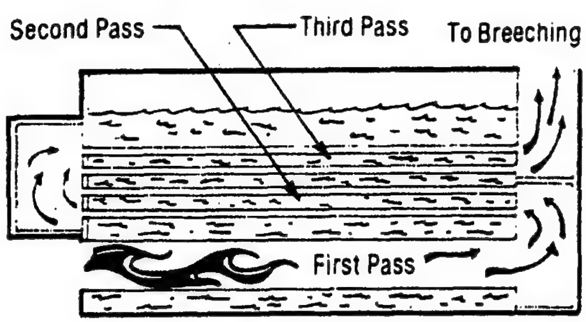


Figure B-6. Boiler Passes.

Source: Dukelow 1983. Used with permission of Kansas State University, Manhattan.

Note that the number of passes does not determine the efficiency of a firetube boiler. Generally, increased passes increase consumption of air blower power due to increased resistance to flow.

Watertube Boilers

The watertube boiler gets its name from the circulation of water through the boiler tubes. The tubes generally connect two cylindrical drums. The higher drum—the steam drum—is half filled with water. The lower drum—the mud drum—is filled completely with water. The lower drum collects any sludge that may develop. The heating of the riser tubes causes a release of steam in the steam drum. A packaged watertube boiler is shown in Figure B-7. Hot water can be generated using the same principle.

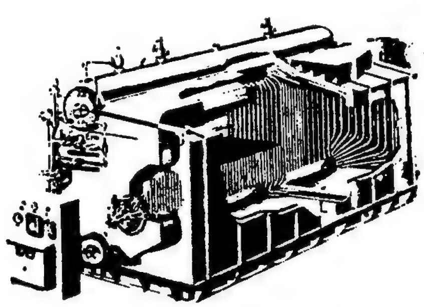


Figure B-7. Watertube Boiler.

Source: Dukelow 1983. Used with permission of Kansas State University, Manhattan.

Watertube boilers are applicable for a wide range of sizes and pressures. Pressures range from 50 to 5,000 psig. Sizes range from 20,000 to 1,000,000 lb/h of steam for industrial watertube boilers. Watertube boilers using solid fuels require greater spacing between the boiler tubes than boilers using liquid and gaseous fuels. This requirement is due to the buildup of ash residue and other particulates on pipes, which reduces air circulation around the pipes. This makes converting a gas- or oil-fired boiler to a coal-firing boiler difficult. Conversion from a coal boiler to a gas or oil boiler is more easily accomplished.

Cast Iron Sectional Boilers

Cast iron sectional boilers are also called watertube cast iron boilers, even though there are no tubes in them. These boilers can be expanded by adding sections. As shown in Figure B-8, the combustion gases flow around the sections that contain water.

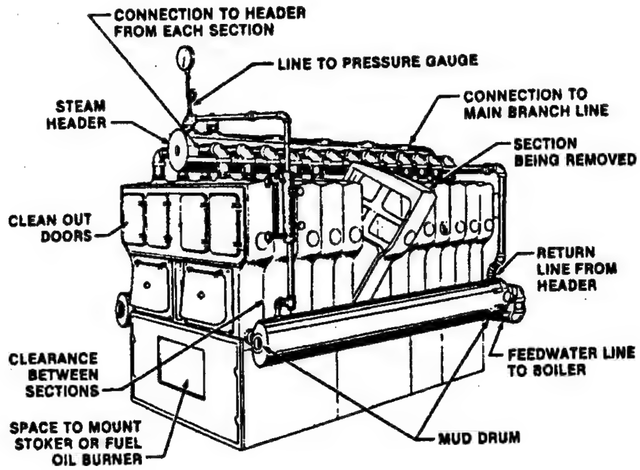


Figure B-8. Cast Iron Sectional Boiler.

Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

Forced Draft Boilers

A forced draft boiler (Figure B-9) consists of a burner and a blower. Air is pushed through the burner wind box.

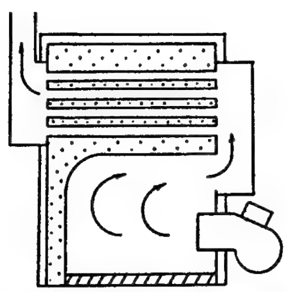


Figure B-9. Forced Draft Boiler.

Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

Natural Draft Boilers

The draft in the natural draft boiler is caused by the difference in weight of the column of flue gases within the stack, and a corresponding column of equal dimensions outside. The intensity of the draft is negative and is expressed in inches of water.

Induced Draft Boilers

A fan is used to pull the air and combustion products through the boiler. The fan is located in an area of the boiler that will allow it to suck particles through the

boiler, not permitting ash, etc., to settle and clog the air passage. If this is not done, the boiler will become dirty inside and inefficient.

Boiler Components

The main components of a boiler are:

- Feedwater system—supplies the water to the boiler.
- Fuel and combustion system—supplies fuel for making heat and provides air for combustion.
- Steam/water system—collects and controls the steam or water.

Each of these components can be further broken down into more specific components. Figure B-10 illustrates the location of various boiler components for a steam boiler. Descriptions of these components and their functions are provided in the pages following the figure.

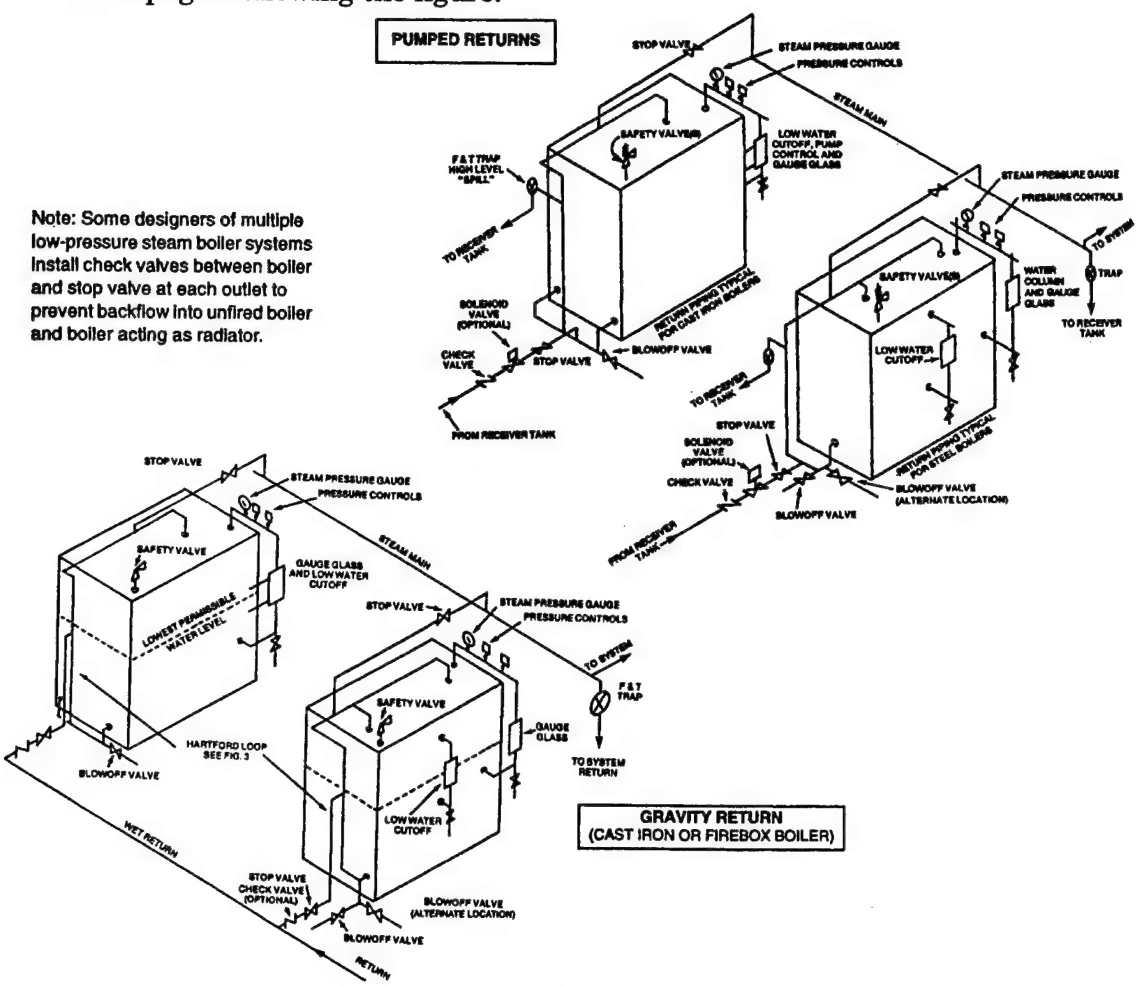


Figure B-10. Steam Boiler.

Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook*.

Components at the Boiler

Safety valve. Considered by many as the most important valve on a boiler, the safety valve will pop open when boiler pressure exceeds the maximum allowable working pressure. The valve is located at the highest part of the steam side of the boiler. No other valves should be located between the safety valve and the boiler. According to the American Society of Mechanical Engineers (ASME), safety valves should be tested every 30 days.

Safety relief valve. Used primarily on water boilers. As with the safety valve, the safety relief valve is an automatic pressure relieving device.

Steam/water pressure gauge. Shows the amount of pressure in the boiler in pounds per square inch (psi). The steam pressure gauge must be viewed easily, and connected to the highest part of the steam side of the boiler.

Water column. Indicates the water level in the boiler. Although the ASME code does not require a water column for all boilers, most steam boilers are equipped with one (Figure B-11).

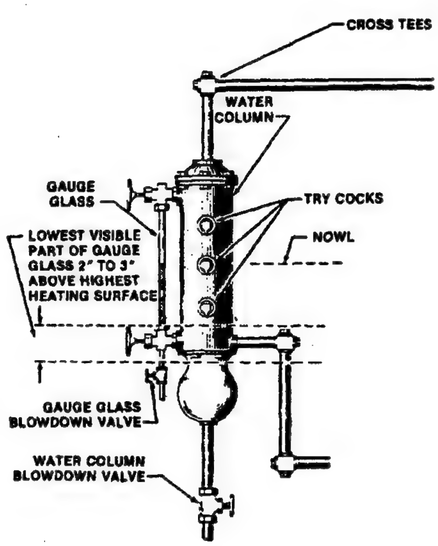


Figure B-11. Water Column and Components.
Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

The water level may be determined using one of two methods. Checking the water level through the gauge glass is the easiest. The gauge glass blowdown valve allows the operator to "blow down" the lines to remove sludge and sediment to

check the water level. This valve needs to be checked every day. The second method involves the use of try cocks. Try cocks are valves that are operated manually. With the try cocks opened and the boiler water level at its normal level, water will spill from the bottom try cock. A steam and water mixture will be discharged from the middle try cock, and the top try cock will disperse steam.

The water column blowdown valve is used to keep the water column and its lines free from sludge and sediment. This valve needs to be checked every day. Bottom blowdown valves are located at the lowest point of the water side of a boiler. Two valves may be used, one a quick-opening, the other a screw type.

Surface blowdown line. Located at the normal operating water level, the surface blowdown line removes the surface impurities, which prevent steam bubbles from breaking through the surface of the water.

Fusible plug. The ASME code requires the fusible plug only on coal-fired boilers; however, they may still be found on gas- and fuel-oil-fired burners. The fusible plug is the boiler's last warning of a low water level. When the water level is low, the tin in the plug melts and allows steam to escape causing a whistling noise to alert the operator.

Boiler vent. A 1/2 or 3/4 in. line with a valve on it coming off the highest part of the steam side of the boiler. The boiler vent must be kept open when filling the boiler with water to prevent the build up of pressure within the boiler. The boiler vent must be kept open when warming up the boiler to allow the air from the steam side to vent. The boiler vent must also be kept open when taking the boiler off-line to prevent a vacuum from forming. Try cocks may be used in the absence of a boiler vent. Safety valves should never be used to vent a boiler.

Pressure control. Located at the highest part of the steam side of the boiler, the pressure control is a switch that turns the burner on or off based on steam pressure.

Feedwater System Components

The feedwater system (Figure B-12) supplies the boiler with water at a certain temperature and pressure.

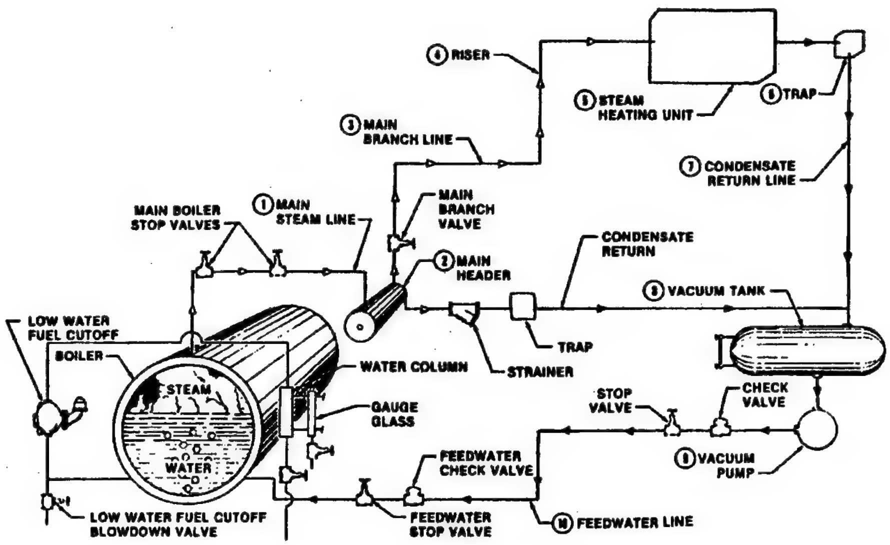


Figure B-12. Feedwater System.

Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

Feedwater stop valve. Permits the flow of water to the boiler when opened. This valve should be located as close to the boiler as possible and is operated manually.

Feedwater check valve. Located between the stop valve and the feedwater pump, this valve allows the water to flow in one direction and prevents water from flowing out of the boiler into the feedwater line. This valve operates automatically.

Vacuum pump. The vacuum pump moves water from the vacuum tank to the boiler. During this process, the vacuum pump creates a vacuum on the return lines, which draws condensate back to the vacuum tank. The pump removes and discharges all air in the water to the atmosphere, and it discharges all the water back to the boiler.

City water makeup. Additional water needed in the system is called makeup water. This water replaces water lost due to leaks or blowing down the boiler. This additional water is added through the city makeup system, shown in Figure B-13. The system can be automatic or manual.

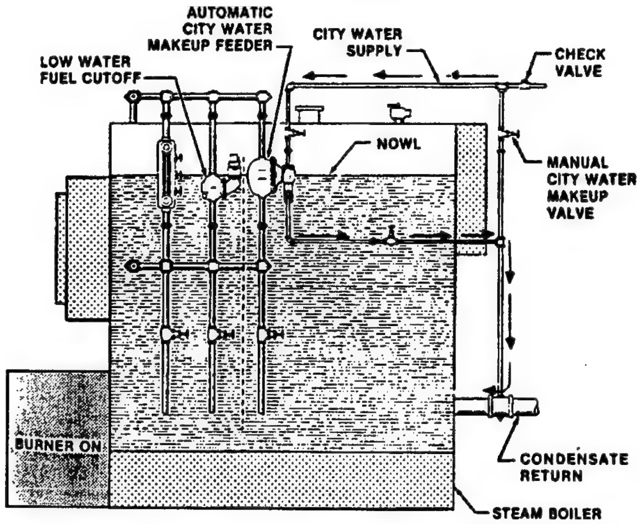


Figure B-13. Makeup System.

Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

Low water fuel cutoff. The ASME code requires low pressure boilers to have a low water fuel cutoff. Located slightly below the normal operating water level, the low water fuel cutoff shuts off the burner when water level is low. This component should be checked daily.

Feedwater regulator. Located at the normal operating water level, the feedwater regulator maintains a constant water level in the boiler by starting and stopping the feedwater pump.

Fuel and Combustion System Components

In the combustion process, fuel is mixed with air, and is burned to produce heat necessary to operate the boiler. The types of fuel commonly used in low pressure boilers are fuel oil, gas, and coal. The factors determining the selection of fuel include the price and availability of the fuel, local pollution codes and regulations, and the boiler design. The combustion components will be discussed according to the different fuel systems.

Fuel oil system.

Fuel oil heaters: Used to heat some grades of oil for ease in pumping and used to heat other oils to allow for burning. There are many separate components necessary for the proper performance of the fuel oil heater.

Fuel oil strainers: The purpose of a strainer in the fuel oil system is to remove foreign matter. It will be necessary to clean these strainers more often when using heavier grades of fuel oil.

Fuel oil pump: The fuel oil pump draws the fuel oil from the fuel oil tank and delivers it to the burner at a controlled pressure.

Fuel oil burner: The fuel oil is delivered to the furnace in a fine spray via the fuel oil burner, providing efficient combustion. There are different types of fuel oil burners.

Gas system. In a gas system, gas burners supply the proper mixture of air and gas to the furnace so complete combustion is achieved. As with the fuel oil system, there are many components necessary to maintain safety and efficiency. Figure B-14 shows the various components of a gas burner system.

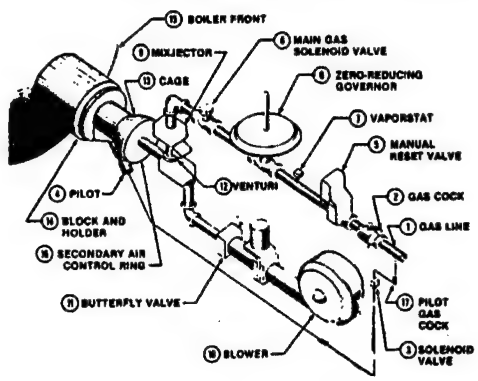


Figure B-14. Gas Burner System.
Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

Gas train: A gas train consists of all components required to provide gas supply to the burner. Each regulator on the gas train must have a separate vent to the outside. Figure B-15 shows the different components of a gas train.

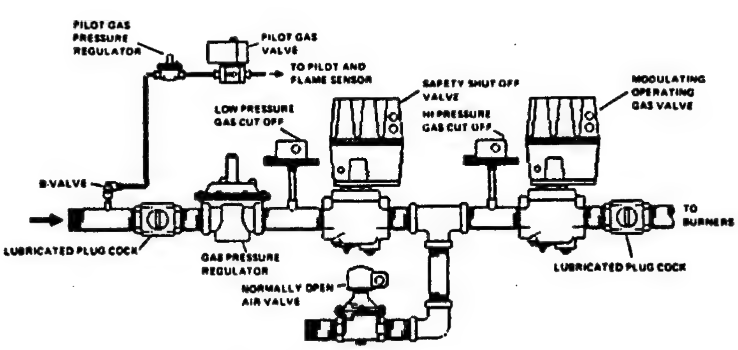


Figure B-15. Gas Train.
Source: Cleaver Brooks. Used with permission.

Coal systems. Coal can be fed using pulverizers, stokers, or by hand firing, which is rarely used anymore.

Stokers: A mechanical coal-feeding device that feeds the coal to the furnace consistently. Use of a stoker also increases efficiency because automatic feeding eliminates the need to open the fire door. The most common type of stoker used in the Army, Air Force, and Navy is the spreader (Figure B-16).

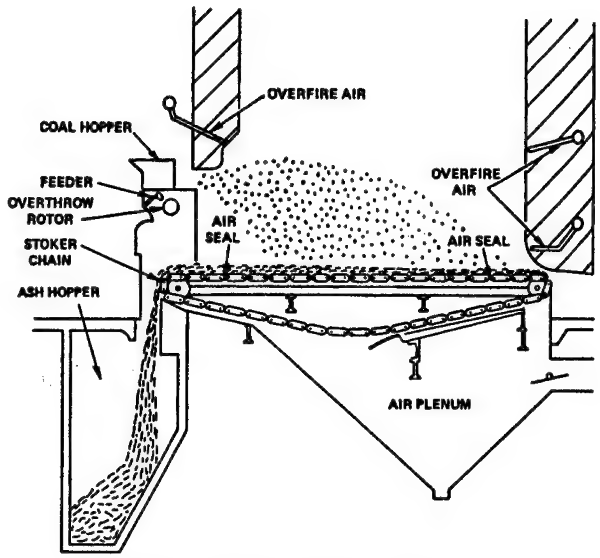


Figure B-16. Spreader Stoker.

Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

Automatic combustion controls. Safety and operational efficiency in combustion is maintained with the proper air to fuel mixture, and by controlling the firing rate of the fuel. Combustion controls regulate the fuel in proportion to steam/water demand, the air supply, and the ratio of air to the fuel supplies. The "ON/OFF" is the most common combustion control, which regulates the burner by the amount of steam pressure/water temperature in the boiler.

Steam/Hot Water Components

Steam is generated within the boiler. It is then piped to areas needing either heat or energy for other industrial applications.

Main steam/water stop valve (Figure B-17). Located on the main steam/water line, the purpose of this valve is to allow for cutting the boiler in on the line and for taking the boiler off-line. This valve should be an outside screw and yoke (OS&Y) valve (Figure B-18). A globe valve should never be used for a main steam/water stop valve. When an OS&Y valve is used, it is visually possible to

tell when the valve is completely open or closed. The valve is open when the stem is up. A globe valve should not be used because it is difficult to know when the valve is completely open or closed. If it is partially open when steam shoots through, the valve will quickly erode, reducing its effectiveness as a stop.

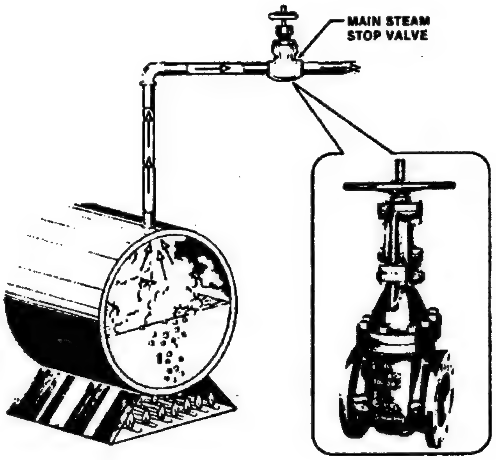


Figure B-17. Main Steam/Water Stop Valve.

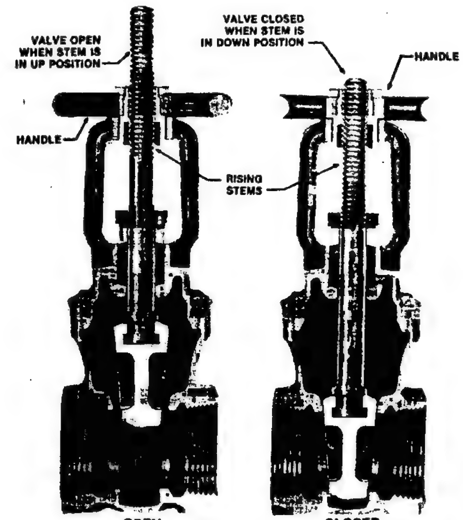


Figure B-18. OS&Y Valve.

Source: Steingrass 1986. Used with permission of American Technical Publishers, Inc.

Steam traps. The purpose of a steam trap is to remove the condensate that forms in a steam line during temperature drops. If the condensate is not removed from the steam line, water hammer will occur. Water hammer can cause pipes to rupture in some cases and disturbing noises in others. Steam traps should be located wherever condensate buildup can occur. These areas are the ends of the main steam branch header, the ends of the main steam branch line, and on each radiator or heat exchanger where steam gives up its heat. Steam strainers should be located in the steam line in front of the steam trap.

Return steam traps are no longer used, but may still be found in older systems. The return steam trap returns the condensate directly to the boiler.

Nonreturn steam traps are used on all low pressure steam systems. The nonreturn steam trap sends the condensate through a vacuum pump to a condensate return tank, which in turn pumps the steam to the boiler. There are three types of nonreturn steam traps. They are the inverted bucket steam trap, the thermostatic steam trap, and the float thermostatic steam trap.

Fuel Burners

Gas and oil are the primary types of fuel burners for packaged boilers. Burners must be able to perform five functions:

1. Deliver fuel to the combustion chamber
2. Deliver air to the combustion chamber
3. Mix the fuel and air
4. Ignite and burn the mixture
5. Remove the products of combustion.

Gas-fired burners. Gas burners are classified according to the pressure available at the gas inlet valve. They can be low pressure (2 to 8 oz per square inch), intermediate (8 oz to 2 psig), or high (2 to 50 psig).

A pilot burner usually is used to ignite gas burners. These pilots can either be continuous or lit each time the burner is started up and shut down after each use.

The gas burner can be modulated to provide satisfactory combustion. This is commonly done by adjusting the air and gas flow simultaneously by:

- using a gas valve and air damper in parallel
- varying gas pressure
- varying air pressure.

Oil-fired burners. The major difference between oil and gas burners is the fact that oil is pumped to the burner by a fuel oil pump. Also, in oil burners the fuel needs to be prepared so that it will burn properly. This preparation is done by atomization and vaporization. Atomization changes the oil into tiny droplets, and vaporization turns these tiny droplets to gas by the heat of the furnace. Oil burners are classified by the means in which the oil is atomized. These are:

- pressure
- steam or compressed air
- rotary.

Oil burners are typically ignited by a high electric voltage spark, or by a high temperature electric heating element.

2 Chemistry of Combustion

Chemical reactions are an important part of the combustion process in delivering energy in the form of heat to boiler surfaces. In fact, the chemical internal energy, which is the energy associated with the destruction and formation of chemical bonds between atoms, will either provide the necessary heat input or will not.

Fuels

Any material that can be burned to release energy is called a fuel. Most familiar fuels consist primarily of hydrogen and carbon, commonly known as hydrocarbon fuels, and they exist in all phases (i.e., coal, gasoline, and natural gas).

The major component of coal is carbon. Coal also contains varying amounts of oxygen, hydrogen, nitrogen, sulfur, moisture, and ash. The difficulty of analyzing coal's mass lies in the variety of its composition from one geographic location to another.

Most liquid hydrocarbon fuels are a mix of numerous hydrocarbons and are distilled from crude oil. The more volatile hydrocarbons vaporize first, forming gasoline. The less volatile fuels obtained during distillation are kerosene, diesel fuel, and fuel oil. The composition of a particular fuel depends on the source of the crude oil as well as on the refinery. Although liquid hydrocarbon fuels are mixtures of many different hydrocarbons, they are usually considered to be a single hydrocarbon for convenience in analysis. For example, gasoline is treated as octane (C_8H_{18}) and diesel fuel as dodecane ($C_{12}H_{26}$). Another common liquid hydrocarbon fuel is methyl alcohol (CH_3OH), also called methanol, which is used in some gasoline blends. The gaseous hydrocarbon fuel natural gas, which is a mixture of methane and smaller amounts of other gases, is sometimes treated as methane (CH_4) for simplicity.

Many types of fuel oils are available for heating and are broadly classified as distillate fuel oils (lighter oils) or residual fuel oils (heavier oils). ASTM has established specifications for fuel oil properties that subdivide the oils into various

grades. Grades number 1 and 2 are distillate fuel oils. Grades 4 and 5 (light), 5 (heavy), and 6 are residual oils. Specifications for the grades are based on required characteristics of fuel oils for use in different type burners. Characteristics that determine grade classification and suitability for a given application are flash point, viscosity, pour point, water and sediment content, carbon residue, ash, distillation qualities, specific gravity, sulfur, carbon hydrogen content, and heating value. Not all of these are included in the ASTM standards.

Combustion

A chemical reaction during which a fuel is oxidized and a large quantity of energy is released is called combustion. Combustion can also be described as the rapid burning of fuel and oxygen that results in the release of heat. Approximately 14 to 15 lb of air is needed to burn a pound of fuel.

Types of Combustion

The three types of combustion are perfect, complete, and incomplete. *Perfect combustion* occurs when all the fuel is burned using only the theoretical amount of air. The theoretical amount of air is the amount of air used to achieve perfect combustion in a laboratory; this would include use of the primary and secondary air, and no excess air. These classifications of air will be explained in the section on efficient combustion. Perfect combustion is seldom, if ever, achieved in a boiler. *Complete combustion* occurs when all the fuel is burned using the minimum amount of air above the theoretical amount of air needed to burn the fuel. Complete combustion is the boiler operator's goal. When complete combustion is achieved, the fuel is burned at the highest combustion efficiency with minimum pollution. *Incomplete combustion* occurs when all the fuel is not burned, resulting in the formation of soot and smoke.

Combustibles

Air is necessary for combustion of fuel. On a mole or volume basis, dry air consists of 20.9 percent oxygen, 78.1 percent nitrogen, 0.9 percent argon, and small amounts of carbon dioxide, helium, neon, and hydrogen. In the analysis of combustion processes, the argon is treated as nitrogen, and the other trace amounts of gases are disregarded. So the oxygen is approximately 21 percent, and nitrogen approximately 79 percent by mole numbers. Pure oxygen O_2 is used as an

oxidizer only in some specialized applications where air cannot be used. Oxygen will support combustion, but it is not a combustible.

A combustible is a material or element that will catch fire and burn when subjected to fire. A combustible will not burn without the introduction of other elements. Oxygen is not easily kindled or excited without the presence of other elements. Nitrogen is not a combustible and will not support the combustion process.

Efficient Combustion

Air used in the combustion process is classified into three types: primary air, secondary air, and excess air. *Primary air* controls the rate of combustion, which determines the amount of fuel that can be burned. *Secondary air* controls combustion efficiency by controlling how completely the fuel is burned. *Excess air* is air supplied to the boiler that is more than the theoretical amount needed to burn the fuel.

When firing a boiler, the operator's goal is to achieve complete combustion. This means burning all fuel using the minimum amount of air. Obtaining complete combustion requires the proper mixture of fuel and air, atomization, and fuel temperature, and enough time to finish the combustion process. Atomization is the breaking of fuel into smaller particles so it will be better exposed to air, which will improve combustion. High firing rates burn the maximum amount of fuel and require more air than low firing rates.

The boiler operator must maintain efficient combustion to minimize the amount of smoke produced. Efficient combustion reduces fuel costs and air pollution. If combustion is not completed before gases come in contact with the cooler surfaces, as the gases cool, they will produce soot and smoke. These will build up and act as an insulator, reducing the amount of heat transfer to the water.

The Combustion Process

Obviously, bringing oxygen into intimate contact with fuel will not start a combustion process. If it did, the whole world would be on fire. The fuel must be brought above its ignition temperature to start the combustion. The ignition temperatures and upper and lower flammability limits of various substances in atmospheric air are listed in Table B-1.

Substance	Molecular symbol	Lower flammability limit[a] %	Upper flammability limit[a] %	Ignition temperature[a] °F	References
Carbon (activated coke)	C			1220	Hartman (1958)
Carbon Monoxide	CO	12.5	74	1128	Scott *et al.* (1948)
Hydrogen	H_2	4.0	75.0	968	Zabetakis (1956)
Methane	CH_4	5.0	15.0	1301	*Gas Engineers Handbook (1965)*
Ethane	C_2H_6	3.0	12.5	968-1166	Trinks (1947)
Propane	C_3H_8	2.1	10.1	871	NFPA (1962)
Butane, n	C_4H_{10}	1.86	8.41	761	NFPA (1962)
Ethylene	C_2H_4	2.75	28.6	914	Scott *et al.* (1948)
Propylene	C_3H_6	2.00	11.1	856	Scott *et al.* (1948)
Acetylene	C_2H_2	2.50	81	763-824	Trinks (1947)
Sulfur	S			374	Hartman (1958)
Hydrogen Sulfide	H_2S	4.3	45.50	558	Scott *et al.* (1948)

Flammability limits adapted from Coward and Jones (1952)

[a] All values corrected to 60 °F, 30 in. Hg, dry

Table B-1. Ignition Temperatures and Flammability Limits

Reprinted with permission from 1997 *ASHRAE Handbook.*

Upper and lower flammability limits are simply the range within which an element or material has the capacity for combustion. In Table B-1, at 60 °F and at 30 in. of mercury (Hg), using methane (CH_4) for an example, combustion is most likely between 5 and 15 percent capacity for combustion. As the percentage of flammability for a given material goes up, the rate of combustion becomes greater.

Many questions about combustion processes can be answered quantitatively. Stoichiometry is the branch of chemistry that deals with the quantitative relationships between elements and compounds in chemical reactions. The atomic theory of matter is basic to stoichiometry. Table B-2 lists oxygen and air requirements for stoichiometric combustion of some pure combustible materials (or components) found in common fuels. For many combustion calculations, only approximate values for theoretical air are necessary. If complete information on the fuel is not available, values from Table B-3 can be used.

			Stoichiometric Oxygen and Air Requirements			
	Molecular		lb/lb Fuel[a]		ft^3/ft^3 Fuel	
Constituent	Symbol	Combustion Reactions	O_2	Air	O_2	Air
Carbon (to CO)	C	$C + 0.5\alpha_2 \rightarrow CO$	1.33	5.75	--	--
Carbon (to CO_2)	C	$C + O_2 \rightarrow CO_2$	2.66	11.51	--	--
Carbon Monoxide	CO	$CO + 0.5\alpha_2 \rightarrow CO_2$	0.57	2.47	0.50	2.39
Hydrogen	H_2	$H2 + 0.5\alpha_2 \rightarrow H_2O$	7.94	34.28	0.50	2.39
Methane	CH_4	$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$	3.99	17.24	2.00	9.57
Ethane	C_2H_6	$C_2H_6 + 3.5O_2 \rightarrow 2CO_2 + 3H_2O$	3.72	16.09	3.50	16.75
Propane	C_3H_8	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	3.63	15.68	5.00	23.95
Butane	C_4H_{10}	$C_4H_{10} + 6.5O_2 = 4CO_2 + 5H_2O$	3.58	15.47	6.50	31.14
--	C_nH_{2n+2}	$C_nH_{2n+2} + (1.5n + 0.5)O_2 \rightarrow nCO_2 + (n + 1)H_2O$	--	--	$1.5n + 0.5$	$7.18n + 2.39$
Ethylene	C_2H_4	$C_2H_4 + 3O_2 \rightarrow 2CO_2 + 2H_2O$	3.42	14.78	3.00	14.38
Propylene	C_3H_6	$C_3H_6 + 4.5O_2 \rightarrow 3CO_2 + 3H_2O$	3.42	14.78	4.50	21.53
--	C_nH_{2n}	$C_nH_{2n} + 1.5nO_2 \rightarrow nCO_2 + nH_2O$	3.42	14.78	$1.50n$	$7.18n$
Acetylene	C_2H_2	$C_2H_2 + 2.5\alpha_2 \rightarrow 2CO_2 + H_2O$	3.07	13.27	2.50	11.96
--	C_nH_{2m}	$C_nH_{2m} + (n + 0.5m)O_2 \rightarrow nCO_2 + mH_2O$	--	--	$n + 0.5m$	$4.78n + 2.39m$
Sulfur (to SO_2)	S	$S + O_2 \rightarrow SO_2$	1.00	4.31	--	--
Sulfur (to SO_3)	S	$S + 1.5O_2 \rightarrow SO_3$	1.50	6.47	--	--
Hydrogen Sulfide	H_2S	$H_2S + 1.5O_2 \rightarrow SO_2 + H_2O$	1.41	6.08	1.50	7.18

[a] Atomic masses: H = 1.008; C = 12.01; O = 16.00; S = 32.06

Table B-2. Stoichiometric Oxygen and Air Requirements for Combustible Materials.

Reprinted with permission from 1997 *ASHRAE Handbook.*

Not having enough air to combine with the components of fuel stoichiometrically will prevent 100 percent oxidation of all the fuel components, and a lower efficiency will result. Figures B-19 and B-20 and Tables B-3 through B-5 show how to determine the right amount of excess air and CO_2 for highest combustion efficiency. From Table B-3, natural gas requires a minimum amount of air (theoretical air = 9.6 lb/lb of fuel) for complete combustion. This amount of air is based on the amount of oxygen molecules needed to combine with the fuel (its stoichiometric equation is found in Table B-2 under methane). The approximate theoretical CO_2 values for stoichiometric combustion of other common types of fuel, as well as CO_2 values for differing amounts of excess air, are given in Table B-4. Desirable amounts of CO_2 depend on the excess air, fuel, firing method, and other considerations. CO_2 is important because it is an indication that carbon has

oxidized. When carbon is oxidized, combustion has taken place; the greater the combustion of carbon, the greater the efficiency of the combustion system.

Type of Fuel	Theoretical Air Required for Combustion
Solid fuels	lb/lb fuel
Anthracite	9.6
Semibituminous	11.2
Bituminous	10.3
Lignite	6.2
Coke	11.2
Liquid fuels	lb/gal fuel
No. 1 fuel oil	12.34 (103)
No. 2 fuel oil	12.70 (106)
No. 5 fuel oil	13.42 (112)
No. 6 fuel oil	13.66 (114)
Gaseous fuels	ft^3/ft^3 fuel
Natural gas	9.6
Butane	31.1
Propane	24.0

Table B-3. Theoretical Amounts of Air Required for Combustion.
Reprinted with permission from 1997 *ASHRAE Handbook.*

Type of Fuel	Theoretical or Maximum CO_2, %	Percent CO_2 at Given Excess Air Values 20%	40%	60%
Gaseous Fuels				
Natural Gas	12.1	9.9	8.4	7.3
Propane Gas (Commercial)	13.9	11.4	9.6	8.4
Butane Gas (Commercial)	14.1	11.6	9.8	8.5
Mixed Gas (Natural and Carbureted Water Gas)	11.2	12.5	10.5	9.1
Carbureted Water Gas	17.2	14.2	12.1	10.6
Coke Oven Gas	11.2	9.2	7.8	6.8
Liquid Fuels				
No. 1 and 2 Fuel Oil	15.0	12.3	10.5	9.1
No. 6 Fuel Oil	16.5	13.6	11.6	10.1
Solid Fuels				
Bituminous Coal	18.2	15.1	12.9	11.3
Anthracite	20.2	16.8	14.4	12.6
Coke	21.0	17.5	15.0	13.0

Table B-4. Theoretical CO_2 Values for Stoichiometric Combustion.
Reprinted with permission from 1997 *ASHRAE Handbook.*

Now that the exact amount of air needed under perfect conditions for complete combustion (which conditions will probably never exist) have been theoretically established, refer to an example analysis to gain a better understanding of Table B-4. Figure B-19 illustrates an actual flue gas analysis performed on a battalion headquarters building, using an electronic combustion analyzer. A low efficiency of 78.7 percent on the initial test is shown. Then the excess air was reduced by 29 percent, raising the efficiency by 2 percent with a final efficiency of 80.7 percent. Using the values for natural gas from Table B-4:

	Theoretical or Maximum CO_2 %	Percent CO_2 at given excess air values		
		20%	40%	60%
Natural Gas	12.1	9.9	8.4	7.3

In the second test, excess air induced is 39 percent at 8.48 CO_2 (8.48 is found from Table B-5 by interpolation). The amount of theoretical air required for combustion of natural gas is found in Table B-3 to be 9.6. If the actual amount used for combustion in the "after adjustment" data of Figure B-19 is used, a value of 6.4 is found. Subtracting 6.4 from 9.6 gives a percent difference of 3.2 between theoretical and actual air used for combustion. The data in Table B-5 is close enough in value to actual testing data that may be used to determine the excess air needed based on percent CO_2 found in flue gases. Note what happens on Table B-5 if excess air goes up; efficiency goes down. If the CO_2 content goes down in the flue gas, less carbon is oxidized, and the efficiency goes down. The two example tests are plotted on Figure B-20.

Figure B-20 appears to give exact values of excess air as correlated to the tests. The example shows that boiler efficiency is a function of temperature as well as O_2, CO_2, and excess air content. This combustion efficiency for gas relates all these elements and presents the total combustion efficiency.

BACHARACH MODEL 300
****
COMBUSTION ANALYZER

ID: 7846

DATE: 26 Jun 90

TIME: 0945 am/pm

FUEL: NATURAL GAS

PRIMRY TEMP(F): 88
STACK TEMP(F): 416

% OXYGEN: 9.1
% EXCESS AIR: 68.0

% CARBON - DIOXIDE: 6.7
PPM CARBON - MONOXIDE: 9

% EFFICIENCY: 78.7
% STACK LOSS: 21.3

TEST PERFORMED BY:

COMMENTS before adjustment

BACHARACH MODEL 300
****
COMBUSTION ANALYZER

ID: 7846

DATE: 26 Jun

TIME: ---------am/pm

FUEL: NATURAL GAS

PRIMRY TEMP(F): 88
STACK TEMP(F): 409

% OXYGEN: 6.4
% EXCESS AIR: 39.[illegible]

% CARBON - DIOXIDE: 8.2
PPM CARBON - MONOXIDE: 44

% EFFICIENCY: 80.7
% STACK LOSS: 19.3

TEST PERFORMED BY:

COMMENTS after adjustment

Figure B-19. Flue Gas Analysis; initial test and after 29 percent reduction in excess air.

CO_2		12.1	11.5	11.0	10.4	9.8	9.2	8.7	8.1	7.5	6.9	6.4	5.8
Excess Air		0	4.5	9.5	15.1	21.3	28.3	36.2	45.0	55.6	67.8	82.2	99.3
Oxygen		0	1	2	3	4	5	6	7	8	9	10	11
	°F												
	300	85.6	85.4	85.2	85.0	84.7	84.5	84.2	83.9	83.5	83.0	82.4	81.7
	350	84.6	84.3	84.1	83.8	83.5	83.2	82.8	82.4	81.9	81.3	80.6	79.8
	400	83.5	83.2	82.9	82.6	82.2	81.8	81.4	80.9	80.3	79.6	78.8	77.8
	450	82.5	82.1	81.8	81.4	81.0	80.5	80.0	79.4	78.7	78.9	77.0	75.9
	500	81.4	81.0	80.6	80.2	79.7	79.1	78.6	77.9	77.1	76.2	75.2	73.9
	550	80.3	79.9	79.4	79.0	78.4	77.8	77.2	76.4	75.5	74.5	73.4	71.9
	600	79.2	78.7	78.2	77.7	77.1	76.4	75.7	74.9	73.9	72.8	71.5	69.9
	650	78.1	77.6	77.1	76.5	75.8	75.1	74.3	73.4	72.3	71.1	69.7	67.9
	700	77.0	76.5	75.9	75.3	74.5	73.7	72.9	71.9	70.7	69.4	67.8	65.9
	750	75.9	75.4	74.7	74.1	73.2	72.4	71.5	70.4	69.1	67.7	66.0	63.9
	800	74.8	74.2	73.5	72.8	71.9	71.0	70.0	68.8	67.5	65.9	64.1	61.9
	850	73.7	73.1	72.3	71.6	70.6	69.7	68.6	67.3	65.9	64.2	62.3	59.9
		LOSS PER PERCENT COMBUSTIBLES											
		2.8	3.0	3.2	3.4	3.7	4.0	4.3	4.6	5.0	5.5	6.1	6.8

Table B-5. Combustion Efficiency Chart for Gas.

Source: *Improving Boiler Efficiency,* Dukelow 1983. Used with permission of Kansas State University, Manhattan.

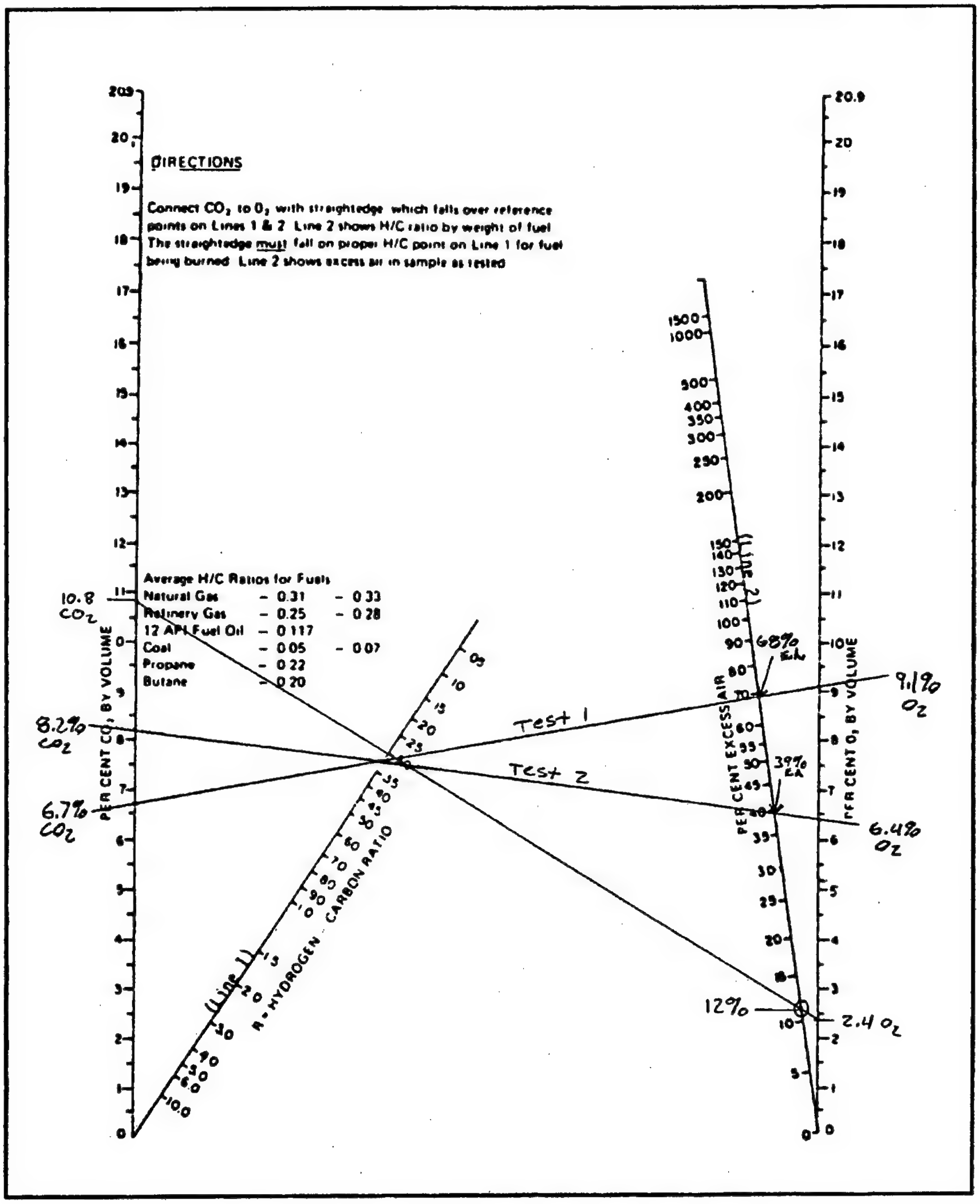


Figure B-20. Properties of Products of Combustion.

Source: *Improving Boiler Efficiency,* Dukelow 1983. Used with permission of Kansas State University, Manhattan.

In calculating the efficiency of boilers, all factors involved must be considered. One problem that makes calculating efficiency difficult is the leaks in the system. If hot water leaks out in the condensate return pipes, it will have to be replaced by makeup water at a much cooler temperature. Also, the makeup air for combustion may have a high moisture content. This means the H_2O is not permitting as much oxidation of H_2 from the fuel, causing incomplete combustion.

Because the needs of the combustion process vary, it is necessary to fluctuate the amount of excess air fed into the combustion process. Mixing of air with fuel may not be sufficiently accomplished to combine or oxidize all fuel components perfectly.

Having considered the obstacles to perfect efficiency in boiler operation, it is apparent we find 100 percent efficiency almost impossible. It is desirable to have all elements of the fuel oxidized by the end of the combustion stage. The theoretical CO_2, ultimate CO_2 or maximum CO_2 concentration attainable in the combustion products of a hydrocarbon fuel with air is the CO_2 concentration obtained when the fuel is completely burned with the theoretical air, or zero excess air. As the carbon-hydrogen ratio of fuel varies, so does the theoretical CO_2 content.

Air Pollution

One of the main constituents of air pollution is that caused by combustion processes. Pollutants may be grouped into four categories:

1. Products of incomplete fuel combustion
   a. Combustible aerosols (solid and liquid), including smoke, soot and organics, and excluding ash
   b. Carbon monoxide (CO)
   c. Gaseous hydrocarbons (HC)
2. Oxides of nitrogen (generally grouped and referred to as NO_x)
   a. Nitric oxide (NO)
   b. Nitrogen dioxide (NO_2)
3. Emissions resulting from fuel contaminants
   a. Sulfur oxides, primarily sulfur dioxide (SO_2) and small amounts of sulfur trioxide (SO_3)
   b. Ash
   c. Trace metals
4. Emissions resulting from additives
   a. Combustion-controlling additives
   b. Other additives.

Emissions of nitrogen oxides and incomplete combustion are directly related to the combustion process and may be minimized by altering the process. During the combustion process, nitrogen oxides form by either thermal fixation (reaction of nitrogen and oxygen at high combustion temperatures) or from fuel nitrogen (oxidation of organic nitrogen in fuel molecules). High excess air and flame

temperature techniques for ensuring complete fuel combustion, and therefore low emissions of incomplete combustion products, tend to promote increased NO_x formation. Emissions of fuel contaminants are related to fuel selection and are slightly affected by the combustion process.

The emission levels of incomplete fuel combustion can be reduced by making sure of adequate excess air, improving mixing of air and fuel (increasing turbulence, improving distribution, and improving liquid fuel atomization), increasing residence time in the hot combustion zone (possibly by decreasing firing rate), increasing combustion zone temperatures (to speed reactions), and avoiding quenching the flame before reactions are completed.

3 Boiler Design

Boiler Sizing

When specifying a boiler, the engineer must determine the pressure rating and the generating capacity or size of the boiler. The boiler horsepower is determined by the following formula:

$$hp = \frac{W(hg - hf)}{34.5\,hfg}$$

where:

hp	=	boiler horsepower
W	=	quantity of dry saturated steam at desired pressure (lb/h)
hg	=	enthalpy of dry saturated water at feedwater temperature
hf	=	enthalpy of saturated water at feedwater temperature
hfg	=	enthalpy of evaporation at 212 °F = 970.3

Note: All enthalpies are in units of Btu/lb.

The actual output that can be expected from a boiler can be determined from the following formula:

$$W = \frac{34.5\,(hp)\,hfg}{hg - hf}$$

Figure B-21 was taken from a manufacturer's catalog. Equations for hp and W are used to determine the operating steam pressure of the boiler. In these equations, the enthalpy may be determined from steam tables. This chart can be used to determine either steam quantity or boiler horsepower when one of these is known (as well as the operating conditions).

Boiler Sizing Example 1

Find the size of a boiler required to generate 4,500 lb of dry saturated steam per hour at 100 psig from 180 °F feedwater. Using Figure B-21:

1. Locate the point of intersection of the psig line and 180°F.
2. Read from left hand scale a value of 32.13 pounds of steam per hour per boiler horsepower.
3. Divide: $$\frac{4{,}500\ lb/h}{32.13\ lb/h\,/\,hp} = 140.05\ hp$$
4. Round off to next standard rating of 150 hp.

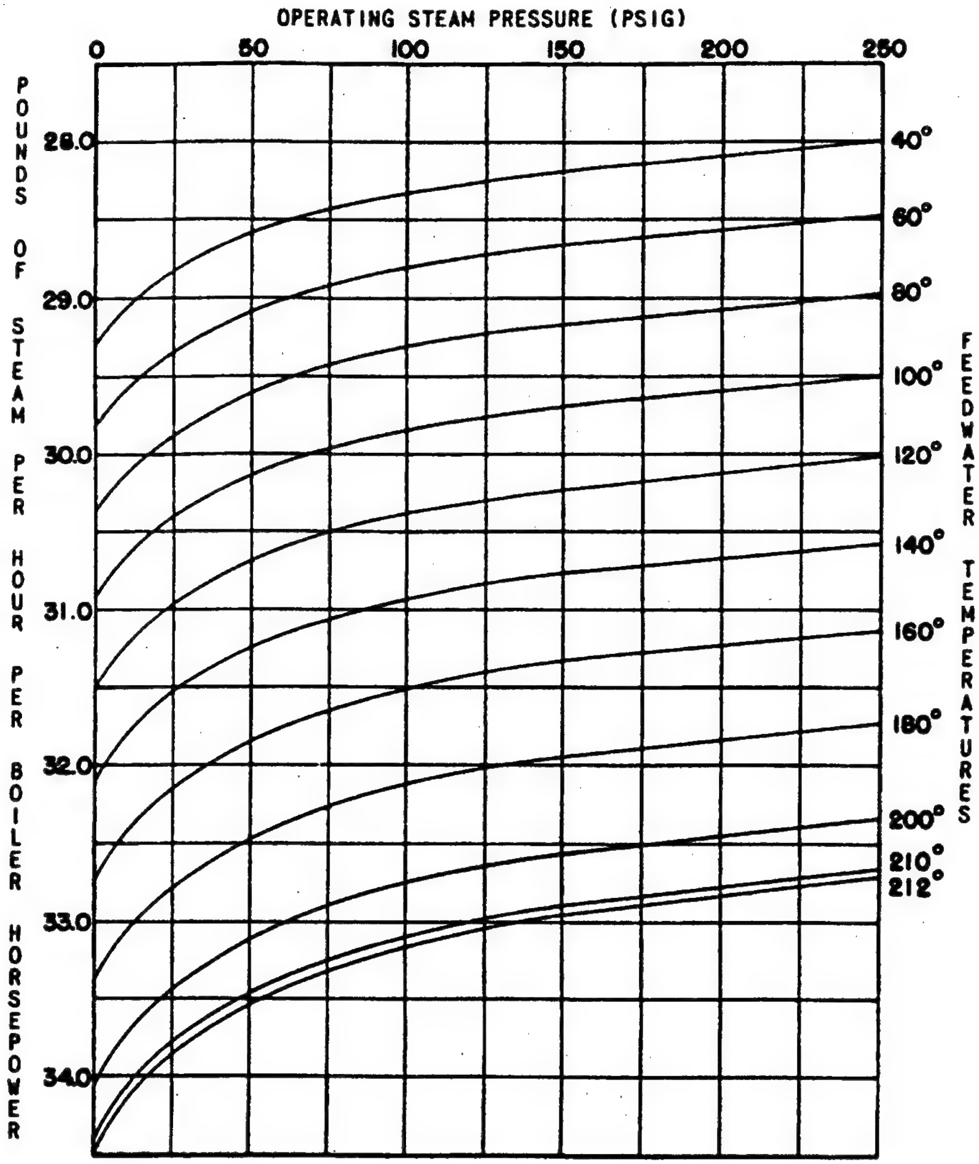


Figure B-21. Pounds of Steam/Hour per Boiler Horsepower vs. Feedwater Temperature and Steam Pressure.

Boilers may also be selected by determining the total MBH output required for the boiler and checking the manufacturer's specifications.

Boiler Sizing Example 2

The total output required for a building is 1,300 MBH. Checking specifications for a cast-iron sectional boiler (see Table B-6), an LGB-14 boiler is selected with a capacity of 1,368.9 MBH.*

Boiler Number*	A.G.A. Input MBH +	A.G.A. Gross Output MBH +	Net I-B-R Ratings**			Net Sq. Ft. Water ***	Boiler HP	Boiler Water Content – Gallons		Approx. Shipping Wt. (Lbs.)	Chimney/ Breeching Size (I.D.) ▲
			Sq. Ft. Steam	Steam MBH +	Water MBH +			Steam (to Waterline)	Water		
LGB-6	650	526.5	1,645	395.0	457.0	3,050	15.7	34.6	54.7	1,725	12"
LGB-7	780	631.8	1,975	474.0	550.0	3,670	18.9	40.3	63.9	2,005	12"
LGB-8	910	737.1	2,305	554.0	642.0	4,280	22.0	46.0	73.0	2,290	14"
LGB-9	1,040	842.4	2,630	632.0	732.0	4,880	25.2	51.9	82.1	2,560	14"
LGB-10	1,170	947.7	2,965	711.0	824.0	5,495	28.3	57.6	91.2	2,800	16"
LGB-11	1,300	1,053.0	3,295	791.0	917.0	6,115	31.4	63.4	100.4	3,105	16"
LGB-12	1,430	1,158.3	3,620	869.0	1,007.0	6,715	34.6	69.1	109.5	3,365	16"
LGB-13	1,560	1,263.6	3,955	949.0	1,099.0	7,325	37.8	74.9	118.6	3,785	16"
LGB-14	1,690	1,368.9	4,310	1,035.0	1,190.0	7,935	40.9	80.7	127.7	4,085	16"
LGB-15	1,820	1,474.2	4,680	1,124.0	1,282.0	8,545	44.0	86.4	136.9	4,355	16"
LGB-16	1,950	1,579.5	5,050	1,212.0	1,374.0	9,160	47.2	92.2	146.0	4,725	17"
LGB-17	2,080	1,684.8	5,410	1,298.0	1,464.0	9,760	50.3	98.0	155.1	4,975	17"
LGB-18	2,210	1,790.1	5,780	1,387.0	1,558.0	10,385	53.3	103.6	164.2	5,270	18"
LGB-19	2,340	1,895.4	6,130	1,471.0	1,649.0	10,995	56.6	109.5	173.4	5,540	18"
LGB-20	2,470	2,000.7	6,470	1,553.0	1,739.0	11,595	59.7	115.3	182.5	5,820	19"
LGB-21	2,600	2,106.0	6,820	1,637.0	1,833.0	12,220	62.9	121.0	191.6	6,080	19"
LGB-22	2,730	2,211.3	7,155	1,717.0	1,924.0	12,825	66.0	126.8	201.2	6,365	19"

Table B-6. Cast Iron Sectional Boiler Specifications.

Boiler Design Checklist

The following items must be considered in the initial design stages to ensure maintenance accessibility and greater operating efficiency.

1. All equipment must be readily accessible. Provide ample room for parts replacement, cleaning, and dismantling.
2. Arrange equipment to take advantage of the most direct runs of pipe.
3. Check and adhere to all local and state codes and regulations.
4. Provide a minimum clearance of 3 ft on all sides and 4 ft above the boiler.
5. Install boiler on a level concrete floor of sufficient strength to support the operating weight of the unit.
6. Install floor drains next to or behind the unit to facilitate flushing the foundation.

* The specific model LGB-14 was selected from Table B-6, which is from a manufacturer's catalog. Different models from other manufacturers may be just as appropriate.

7. Provide ample ventilation. According to one manufacturer, two ventilation openings to the exterior of the building will provide a positive movement of air. These openings should be louvered and filtered to protect against the weather.
8. Provide isolation pads to prevent vibration.

4 Acceptance Testing

Boiler Clearances

Many code books used in engineering have been published for the safety of human life, property, and public welfare. Some of the codes available for limitations on boiler clearances are: Life Safety Code (National Fire Protection Association); Uniform Building Code (International Conference of Building Officials); and Uniform Mechanical Code (International Association of Plumbing and Mechanical Officials). For a thorough breakdown of the limitations on boiler clearances, the Uniform Mechanical Code (UMC) is used. As set forth in the 1985 edition of the UMC, safety requirements are as follows: "All boilers and pressure vessels, and the installation thereof, shall conform to minimum requirements for safety from structural and mechanical failure and excessive pressures, established by the building official in accordance with nationally recognized standards."

The safety requirements include such items as controls, gages, and stack dampers, and integrateable welding by approved welders in conformity with nationally recognized standards. The controls must be approved by an approved testing agency, and provide electrical controls that are suitable for installation in their environment. Gages aid in regulating the safety of boilers by providing pressure measurements and a water level glass for steam boilers. A pressure gage with a temperature indicator on water boilers is necessary.

Section 2114 of the UMC states with regard to clearance for access: "when boilers are installed or replaced, clearance shall be provided to allow access for inspection, maintenance, and repair, and passageways shall have an unobstructed width of no less than 18 in. Clearance for repair and cleaning may be provided through a door or access panel into another area, provided the opening is of sufficient size. Power boilers having a steam generating capacity in excess of 5,000 lb per hour or having a heating surface in excess of 1,000 sq ft or input in excess of 5,000,000 Btu/h shall have a minimum clearance of 7 ft from the top of the boiler to the ceiling."

There are other important safety considerations when installing a boiler. Floors must be constructed of a noncombustible material unless the boilers are listed for mounting on combustible flooring. Boilers must be anchored securely to the structure and be mounted on a level base capable of supporting and distributing the weight contained thereon.

As with all engineering and design problems, each problem has its own solution unique to its own environment and circumstance. The codes found in most books will suffice for safety requirements and maintenance. It is still possible, however, to find that all code requirements are met and, yet, something is lacking. Innovative thinking should be used cautiously so as to comply with all safety standards and still accomplish the task at hand.

Boiler Flue Gas Venting

In venting gases produced in the combustion processes of the various types of boilers (oil burning, gas fired, and multi-fuel), it is important to understand the many ways in which codes effect the design and specifications of flue gas venting systems. A few definitions will be helpful in the following discussion on flue gas vents and vent connectors:

Vent: A listed factory-made vent pipe and vent fitting for conveying flue gases to the outside atmosphere.

Type B Gas Vent: A factory-made gas vent listed by a nationally recognized testing agency for venting listed or approved appliances equipped to burn only gas.

Type L Vent: A venting system consisting of listed vent piping and fittings for use of oil-burning appliances listed for use with Type L or with listed gas appliances.

Vent Connector, Gas: That portion of a gas-venting system that connects a listed gas appliance to a gas vent.

Boilers of all kinds must be connected properly to a chimney or vent. A boiler may make direct use of the flue gas vent only if one boiler is in the system. If multiple boilers are used then a multiple appliance venting system may be used. If two or more oil- or gas-burning appliances are connected to one common venting system as shown in Figure B-22, they may be vented into the same system serving liquid-fuel-fired appliances, provided: (1) the gas appliances are each

equipped with a safety shut off device and (2) each oil appliance is equipped with a primary safety control. The rule of thumb, however, is that gas vents shall be insulated in accordance with the terms of their listings and the manufacturer's instructions.

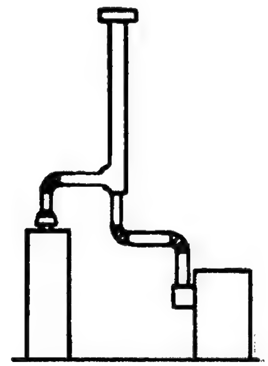


Figure B-22. Boiler Venting.

Table B-7 shows some of the vent types available for various boilers. Venting systems must comply with the following requirements from NFPA 54 and 31:

A single or common gas vent shall be allowed in multi-story installation to vent gas-fueled Category I equipment that is located on more than one floor, under the condition that it is designed and installed under approved methods.

Two or more connectors cannot enter a common venting system unless the inlets are offset in such a way that no portion of any inlet is opposite the other inlets. The smaller connector shall enter at the highest level consistent with the available head room or clearance to combustible material.

When two or more appliances are connected to one venting system, the venting system area must not be less than the area of the largest vent connector plus 50 percent of the areas of the additional vent connectors (NFPA 54, Appendix G).

Each vent connector of a multiple venting system must have the greatest possible rise between the headroom available from the draft hood outlet, the barometric damper or the flue collar, or the point of interconnection to a manifold, to the common vent.

COLUMN I TYPE B, GAS Round or Oval	COLUMN II TYPE BW GAS	COLUMN III TYPE L	COLUMN IV PLASTIC PIPE
All listed gas appliances with draft hoods such as: 1. Central furnaces 2. Floor furnaces 3. Heating boilers 4. Ranges and ovens 5. Recessed wall furnaces (above wall section) 6. Room and unit heaters 7. Water heaters	1. Gas-burning wall heaters listed for use with Type BW vents	1. Oil burning appliances listed for use with Type L vents 2. Gas appliances as shown in first column	1. Condensing appliances listed for use with a specific plastic pipe recommended and identified in the manufacturer's installation instructions

Table B-7. Vent Types.

Based on *Fire Protection Handbook,* 18th Ed., 1997.

It is also important to make sure the venting system is constructed in a way that a positive flow, adequate to convey all combustion products to the outside atmosphere, is produced. It may be tempting to put a vent in a plenum or through an air duct to utilize the heat, but this must not be done. Dangerous gas could seep through and endanger persons in the occupied space.

Some additional codes on connectors are:

- Connectors serving gravity-vent-type appliances shall not be connected to a vent system served by a power exhaust unless the connection is made on the negative side. A gravity vent is operated by the push or upward force on hot air (see Figure B-22). This force is caused by the downward convections of colder, more dense air pulled down by gravity. The hot air continues its acceleration up and out of the vents. If a fan is placed before the outlet of the connection into the common vent, the positive pressure may overcome the force of gravity, and push the exhaust gases back into the occupied space, which is extremely dangerous.
- All connectors shall be as short and straight as possible.
- An appliance shall be located as close as practical to the venting system.
- Connectors shall not be concealed by building construction; however, Type B and L materials may be enclosed following inspection if they meet provisions of section 915 b 2H of the Uniform Mechanical Code.
- Vent connectors shall not pass through any ceiling, floor, fire wall, or partition. A single wall metal pipe connector shall not pass through any interior wall.

- Connectors shall be securely supported, and joints fastened with sheet metal screws, rivets, or other approved means.

Boiler Piping

Boiler or steam piping differs from other systems because it usually carries three fluids: steam, water, and air. Steam systems are classified according to piping arrangement, pressure conditions, and method of returning condensate to boiler. All applicable codes and regulations should be checked to determine acceptable piping practice for the particular application. Codes may dictate piping design, limit the steam pressure, or qualify the selection of equipment.

Two piping arrangements are generally used to suit their own purposes. One of these is the one-pipe system, which uses a single pipe to supply steam and return condensate. Ordinarily, there is one connection at the heating unit for both supply and return. A two-pipe steam system is more commonly used in air-conditioning, heating, and ventilating applications.

Piping arrangements are further classified with respect to condensate return connections to the boiler and direction of flow in the risers:

1. Condensate return to boiler (see Figure B-25)
   a. Dry-return: condensate enters boiler above water line
   b. Wet-return: condensate enters boiler below water line
2. Steam flow in riser
   a. Up-feed: steam flows up riser
   b. Down-feed: steam flows down riser.

Steam piping systems are normally divided into five classifications: high pressure, medium pressure, low pressure, vapor, and vacuum systems. The following are pressure ranges for the five systems:

High Pressure:	100 psig and above
Medium Pressure:	15 to 100 psig
Low Pressure:	0 to 15 psig
Vapor:	Vacuum to 15 psig
Vacuum:	Vacuum to 15 psig

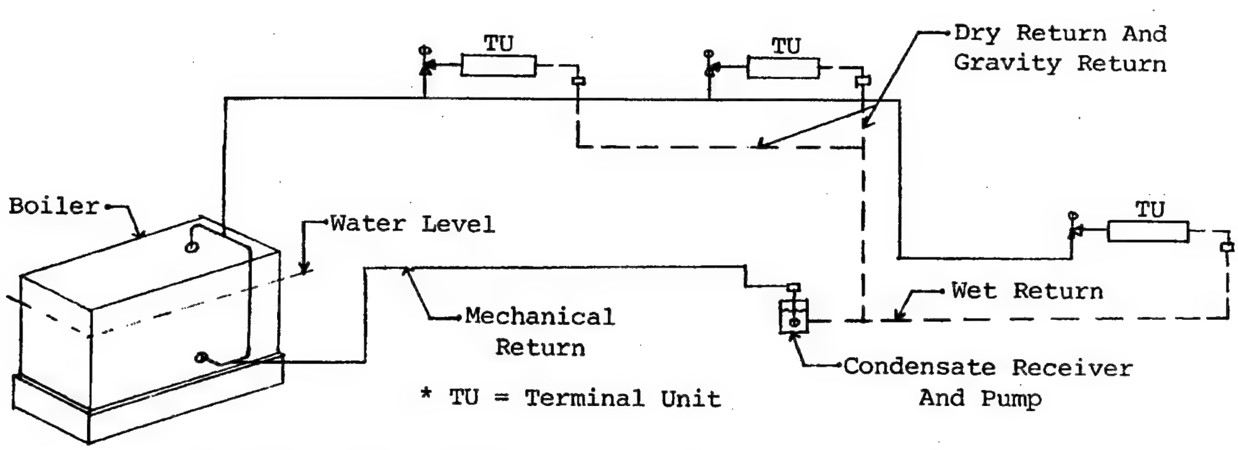


Figure B-23. Dry Return, Wet Return, Gravity Return, Mechanical Return.

Vapor and vacuum systems are identical except the vapor system does not have a vacuum pump as a vacuum system does.

Piping systems are further identified by the type of condensate return piping from the terminal units to the boiler. In common use are two arrangements, gravity and mechanical return. When all the units are located above the boiler or condensate receiver water line, the system is described as a gravity return since the condensate returns to the boiler by gravity (see Figure B-23). If traps or pumps are used to aid the return of condensate to the boiler, the system is classified as a mechanical return system (see Figure B-23).

Water Treatment

A wide range of treatment procedures can be used for boiler waters. In any particular case, the method selected must depend on the composition of the makeup water, the operating pressure of the boiler, the makeup rate, and many other considerations. All makeup water or steam systems should be treated. In acceptance testing, treatment should be done only if available. Refer to Corps of Engineers Guide Specifications for appropriate water treatment provisions.

Efficiency Performance Methods

The performance of a boiler relates directly to its ability to transfer heat from the fuel to the water while meeting operational specifications. A boiler's performance will include all aspects of its operation. Boiler efficiency and operating capacity are basic elements of boiler performance.

Performance specifications include the operating capacity and the factors for adjusting that capacity, steam pressure, boiler water quality, boiler temperatures, boiler pressures, boiler drafts and draft losses, flue gas analysis, fuel analysis, and fuel burned. Other performance specifications indicate the fan power requirements (boiler flue gas temperatures and draft losses) and the fuel supply assumptions. Fuels will vary in their energy content per cubic foot. Higher octane gases will have a greater energy content than those with a lower energy content. Coal varies significantly in energy content and in its capacity to combust. The fuel supply assumptions will take these factors into account.

From the performance specifications, a calculated efficiency may be obtained. Boiler efficiency is a percentage of the ratio of heat supplied to the boiler and the heat absorbed by the boiler water. Two methods of calculating the efficiency of a boiler are acceptable. They are known as the *input/output method* and *heat loss method.* These methods are very detailed and not necessary as an acceptance testing criteria. The primary test required for acceptance testing is the flue gas temperature.

Boiler System Acceptance Testing Checklist

Before an individual or acceptance testing team begins work, an inspection of the entire system should be made to confirm that all components of the system are ready to function. The following is a checklist to follow during acceptance testing.

BOILER SYSTEM ACCEPTANCE TESTING CHECKLIST

PROJECT:______________________________

LOCATION:______________________________

NAME:______________________________

A. Gas-Firing	Correct yes	Correct no	Date Checked
1. Condition and cleanliness of gas injection orifices			
2. Cleanliness and operation of filter and moisture traps			
3. Condition of burner refractory (loose or cracked)			
4. Condition and operation of air dampers (operable)			
5. Flame scanner operational			
6. Pilot ignition set			
7. Ignition time main flame			
8. Pilot flame out time. Main flame out			
9. Operating temperature a. Inlet b. Outlet			
10. Operating pressure			
11. Combustion air adequate			
12. Gas piping leak tested			
13. Gas train components vented			
14. Gas meter reading			
15. Confirm start-up sequence			

B. Oil-Firing	Correct yes	Correct no	Date Checked
1. Cleanliness of oil strainer			
2. Condition of burner throat refractory (loose or cracked)			

C. Combustion Controls	Correct yes	Correct no	Date Checked
1. Cleanliness and proper movement of fuel valves			
2. Excessive "play" in control linkages or air dampers			
3. Adequate pressure to all pressure regulators			
4. Unnecessary cycling of firing			
5. Proper operation of all safety interlocks and boiler trip circuits, i.e. low pressure, high pressure and low gas pressure			

D. Flame Appearance	Correct yes	no	Date Checked
1. Oil & pulverized flames - short, bright, crisp and highly turbulent			
2. Gas flames--blue, slightly streaked or nearly invisible			

E. Flue Material Type and Connection	Correct yes	no	Date Checked
1. Gas flue - type B or IC (replaces type L)			
2. Oil flue - type IC (replaces type L)			
3. Gas-oil - type IC (replaces type L)			
4. Flue installed per listing			
5. Location and size of makeup air			
6. Do exhaust fans affect flue performance?			
7. Does stack have cap?			
8. Is single wall breeching installed?			

F. Boiler	Correct yes	no	Date Checked
1. Combustible floor - boiler approved for combustible floor			
2. 18 in. unobstructed clearance around all sides of boiler			
3. Boiler > 5,000 BtuH - minimum clearance of 7 ft from the top of boiler to ceiling			
4. Make-up water system installed			
5. Make-up water controls set			
6. Feed water auxiliaries operational			
7. Feed water treatment in place			
8. Treatment system discussed with user			
9. Boiler flushed and clean			
10. Pressure relief operational			
11. Operating pressure			
12. Water level control tested			
13. Installation checked and approved by manufacturer			
14. Combustion test complete and results submitted			
15. Pressure relief valve matches boiler capacity			

G. Flue Gas Temperature	Correct yes	no	Date Checked
1. Temperature - actual vs. recommended			

Glossary

ANODE: The positively charged electrode toward which current flows.

ASTM: American Society for Testing and Materials.

ASME: American Society of Mechanical Engineers.

BLOW DOWN: Removal of a portion of boiler water for the purpose of reducing concentration, or to discharge sludge.

BOILER PASSES: The number of passes for a boiler refers to the number of horizontal runs the flue gases take between the furnace and the flue gas outlet.

BURNER WINDBOX: A plenum chamber around a burner in which an air pressure is maintained to ensure proper distribution and discharge of secondary air.

CATHODE: The negative electrode from which current flows.

COMBUSTION: The rapid chemical combination of oxygen with the combustible elements of a fuel resulting in the production of heat.

COMBUSTION CHAMBER: An enclosed space provided for the combustion of fuel.

CONCENTRATION: The strength or density of a solution.

CRUDE OIL: Unrefined oil. When an oil rig first strikes oil underground, the oil as extracted from the ground is in its crude form.

EXCESS AIR: The amount of air supplied to the boiler that is greater than the amount of theoretical air needed to burn the fuel.

GAGE GLASS: The transparent part of a water gage assembly connected directly or through a water column to the boiler, below and above the water line to indicate the water level in a boiler.

HOT WATER-HIGH PRESSURE: A water heating boiler operating at pressures exceeding 160 psi or temperatures above 250 °F.

HOT WATER-LOW PRESSURE: A boiler furnishing hot water at pressures not exceeding 160 psi and temperatures less than 250 °F.

INCHES WATER GAGE: The usual term for expressing a measurement of relatively low pressure or differential by means of a U-tube manometer. One inch w.g. equals 5.2 lb per square foot or 0.036 lb per square inch.

MANIFOLD: A pipe or header for collecting a fluid from or the distributing of a fluid to a number of pipes or tubes.

MECHANICAL STOKER: A device that feeds a solid fuel into a combustion chamber.

MUD DRUM: The lower drum of a watertube boiler in which steam system sediments settle into, and is completely filled with water.

OS&Y VALVE: Outside screw and yoke valve.

OXIDATION: Chemical combination with oxygen.

PRESSURE REDUCING VALVE: A pressure reducing valve for a single temperature system reduces tank pressure to 18 psig. For two temperature systems and systems designed to change from direct to reverse acting, a change in supply pressure will provide a choice of 13 or 18 psig.

PRIMARY AIR: Air that is needed to mix with the fuel. It atomizes and controls the amount of fuel oil capable of being burned.

RETURN CONDENSATE: Condensed water resulting from the removal of latent heat from steam.

SCALING: The formation of deposits in a boiler caused by the minerals in the boiler water.

SECONDARY AIR: Air used in controlling how efficiently the fuel is burned. It is air that diffuses into the flame from the atmosphere.

SLUDGE: The sediment in a steam boiler.

SOOT: Unburned particles of carbon derived from hydrocarbons.

STEAM: The vapor phase of water substantially unmixed with other gases.

STEAM DRUM: The higher drum of a watertube boiler used to contain steam, and is half filled with water.

UMC: Uniform Mechanical Code.

VENTING COLLAR: Outlet opening of an appliance provided for connecting the vent system.

VISCOSITY: The measure of the internal friction of a fluid or its resistance to flow.

WATER HAMMER: The hammering sound caused in a pipe containing condensate or water when live steam is passed through it.

Bibliography

American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), *Fundamentals Handbook,* Atlanta, Georgia, 1997.

ASHRAE, *Systems and Equipment Handbook,* 1996.

Dukelow, S. G., *Improving Boiler Efficiency*, Cooperative Extension Service, Kansas State University, Manhattan, Kansas, 1983.

Glassman, Irving, *Combustion*, Academic Press, New York, New York, 1977.

National Fire Protection Association, *National Fire Code, Volume 3*, Quincy, Massachusetts, 1985.

Steingrass, Frederic M., *Low Pressure Boilers. second edition*, American Technical Publishers, Inc., Homewood, Illinois, 1986.

Strehlow, Roger A., *Fundamentals of Combustion*, Robert E. Krieger Publishing Co. Inc., Huntington, New York, 1979.

Tipper, Minkoff, *Chemistry of Combustion*, Butterworth Inc., Washington, DC, 1962.

Uniform Mechanical Code, Whittier, California 1985.

Appendix C: Chiller Systems

Principles, Applications, and Acceptance Testing

by
Dahtzen Chu
Energy Branch, Facilities Division
U.S. Army Construction Engineering Research Laboratory

and

Charles L. Burton
Stacy Campbell
Alison J. Pacheco
Leland V. Speirs
Department of Architectural Engineering and Construction Science
Kansas State University

Contents

1 Introduction

Chillers are devices that remove the heat that is gained by a recirculating chilled water system as it cools a building. Two types of refrigeration cycles can be used by chillers: either the compression cycle or the absorption cycle.

The term "chiller" is used in connection with a complete chiller package, which includes the following: compressor, condenser, evaporator, internal piping, and controls. "Chiller" is also used when all these components are used with a cooling tower.

Compression Cycle

Liquid refrigerant, at a relatively high pressure and temperature, flows through a restriction called the flow control device or expansion valve. The flow control device separates the high-pressure side of the system from the low-pressure side. It acts as a pressure reducing valve because the pressure of the liquid flowing through it is lowered, and only a small portion of the refrigerant flows through the valve into the evaporator.

The refrigerant that flows through the evaporator is vaporized by the heat flowing through the walls of the evaporator. After leaving the evaporator, the refrigerant is a gas at a low temperature and pressure. To be able to use it again to achieve the refrigerating effect, it must be brought back to a high-pressure liquid. Refrigerant flows from the evaporator to a compressor where the pressure is increased. Compressing the gas also increases the temperature. The refrigerant travels to a condenser after leaving the compressor and flows through one circuit in the condenser. In the other circuit, a cooling fluid (either air or water) flows at a temperature lower than the refrigerant. Heat transfers from the refrigerant to the cooling fluid, and the refrigerant condenses to a liquid. Figure C-1 shows the compression cycle.

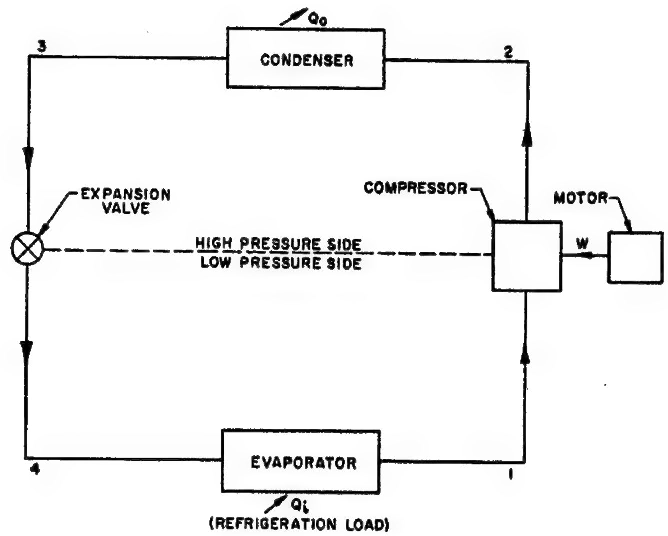


Figure C-1. Compression Refrigeration Cycle.

Reprinted with permission from 1997 *ASHRAE Handbook.*

Centrifugal Compressors

Centrifugal compressors are variable displacement machines that provide pressure through the action of rotating impellers. The fundamental design of the centrifugal compressor is similar to that of a centrifugal pump (Figure C-2).

Refrigerant vapor enters the compressor through suction passages, and passes into the impeller. The impellers increase the velocity of the vapor. The velocity energy resulting from this increase is converted to a pressure increase. Centrifugal compressors are designated by their number of stages, with one stage for each impeller.

Reciprocating Compressors

The definition of reciprocating is a back and forth motion in a straight line. Reciprocating compressors are positive displacement machines that provide compression through the action of a piston squeezing refrigerant in a cylinder. Construction is similar to the reciprocating engine of a vehicle, with pistons, cylinders, valves, connecting rods, and crankshaft (Figure C-3).

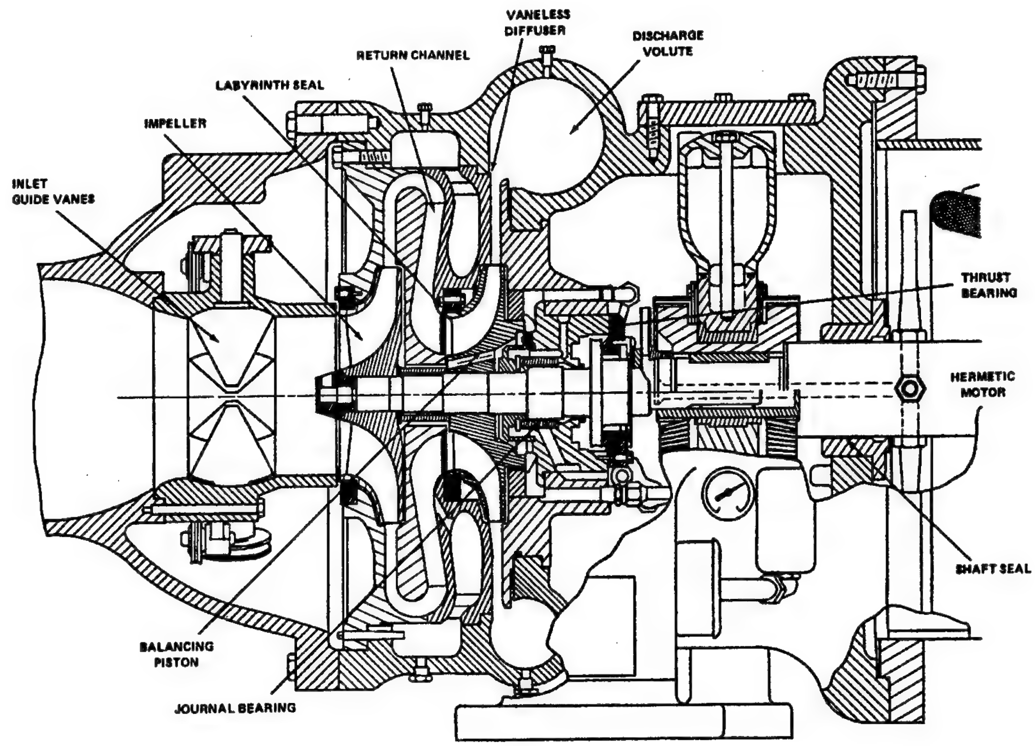


Figure C-2. Centrifugal Compressor.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

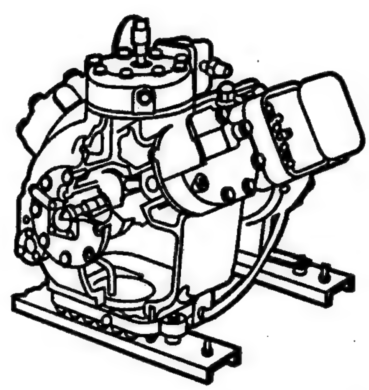


Figure C-3. Reciprocating Compressor.

Carrier Corporation, Syracuse, NY. Used with permission.

As the piston moves out of the cylinder, refrigerant vapor is drawn in. As the piston moves in, the refrigerant is compressed. In most cases, the suction and discharge valves are either thin plates or reeds that will open and close easily and quickly.

The reciprocating compressor is the most widely used type, available in sizes from fractional horsepower and tonnage up to a few hundred tons.

Screw Compressors

Screw compressors can be used in the compression cycle in a complete chiller package. They come in two main types: single screw and twin screw.

Single screw. The single screw compressor (Figure C-4) consists of a single cylindrical main rotor that works with a pair of gate rotors. The compressor is driven through the main rotor shaft and the gate rotors follow. Refrigerant vapor enters the suction chamber. As the main rotor turns, vapor is trapped in the space formed by the three sides of the flutes, casing, and gate rotor tooth. As rotation continues, the flute volume decreases and compression occurs as illustrated in Figure C-5.

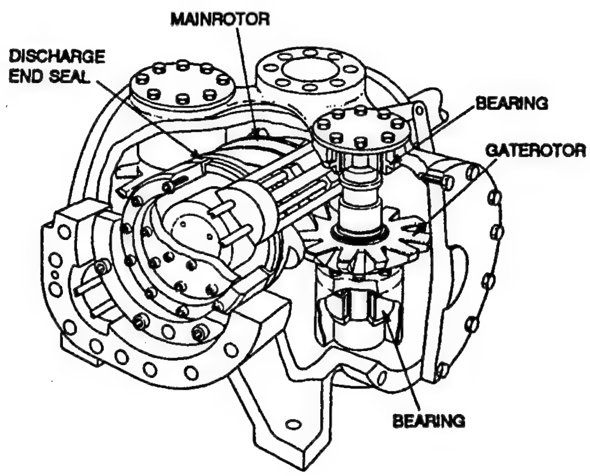


Figure C-4. Single-Screw Compressors.

Reprinted with permission from 1996 *ASHRAE HVAC Systems and Equipment Handbook.*

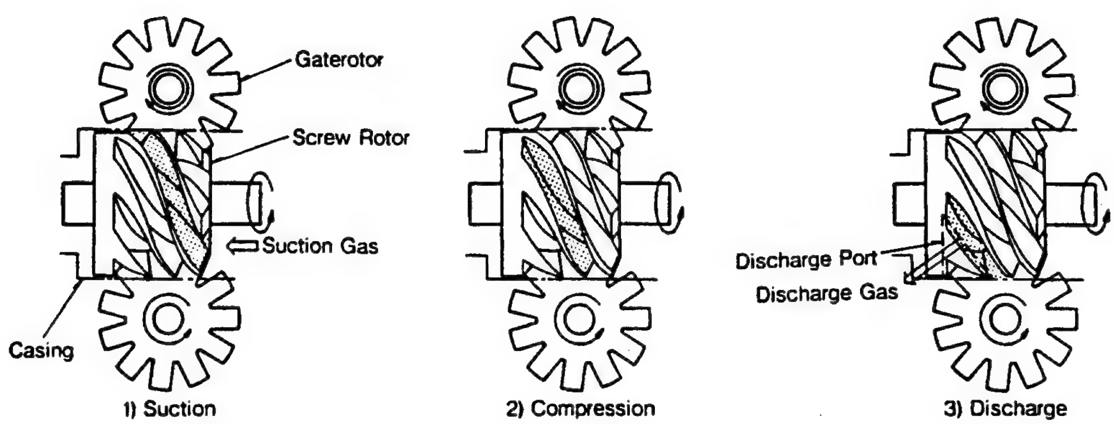


Figure C-5. Single Screw Compressor Sequence of Operation.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

Twin screw. The twin screw compressor (Figure C-6) consists of two mating helical grooved rotors, a male and a female, in a stationary housing with inlet and outlet ports. As the two rotors (screws) rotate, the volume between the screws decreases toward the discharge end, and the vapor is compressed.

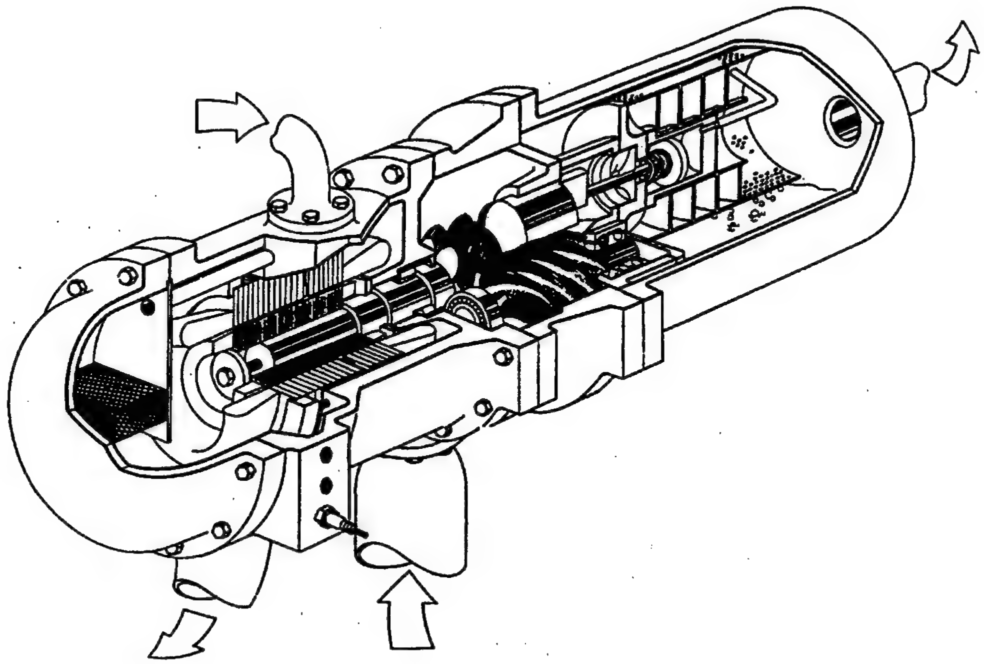


Figure C-6. Twin Screw Compressor.
Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

Scroll Compressors

The scroll compressor is another type of compressor primarily used in residential and automotive air-conditioning.

Absorption Cycle

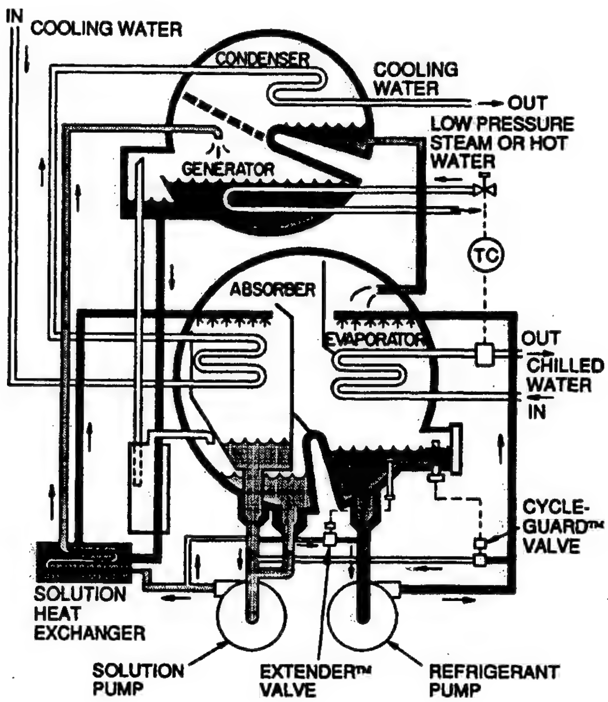


Figure C-7. Absorption Cycle.

Source: Carrier Corporation, Syracuse, NY. Used with permission.

Because absorption processes are often not well understood, a detailed discussion of the absorption cycle will be provided here. Absorption is defined as "the taking up of matter or energy by penetration into an absorbing medium, so that the absorbed matter or energy apparently disappears" (Encyclopedia Americana 1982).

In chemical processes, absorption refers to the solution of a gas in a liquid, the solution being obtained by the washing or intimate contact (scrubbing) of a gas mixture with the liquid. In the ideal situation, an equilibrium is attained, and there is a definite relationship between the concentrations of the gas phase and the liquid phase of the absorbed component.

Gas-liquid absorption is accomplished in vertical counter-current flow patterns through packed, plate, or spray towers. The packed tower is a shell filled with specifically shaped packing materials. Plate towers contain plates at various heights within the tower. In spray towers, the liquid surface is increased by forcing it through spray nozzles to form many tiny droplets that fall through the rising gas stream.

In the absorption system, the low-pressure vapor is absorbed and transported to the high-pressure side in a solution. Once it is on the high-pressure side, it is recovered as a high-pressure liquid refrigerant. In connection with the low side is an evaporator and an absorber. The constituent parts of the high side are the concentrator and the condenser. Excluding energy consumption and efficiency, the absorption process produces the same results as the compression cycle.

Fluids of the Absorption Cycle

Absorption machines use distilled water as a refrigerant. Water is a stable compound having a 1000 Btu per pound of latent heat of vaporization.

Large amounts of water are readily absorbed and separated from the absorbent solution. In the evaporator of an absorption machine, water usually boils at 40 °F, and a pressure equivalent to 1/100 of normal atmospheric pressure. It is also a nontoxic and low cost refrigerant.

The second fluid used in an absorption machine is an absorbent, which is lithium bromide salt in a water solution. The lithium bromide solution is confined to the absorber-concentrator sections of the machine and is the transporter of refrigerant from the low- to the high-pressure side.

Evaporator (low-pressure side)

As with the compression refrigeration cycle, high-pressure liquid refrigerant is passed from the condenser through an orifice (expansion valve) into the lower pressure evaporator. System water having a temperature of about 54 °F enters the evaporator tubes to be chilled while the refrigerant's temperature is at about 40 °F. Because the temperature of the water is higher than that of the refrigerant, heat transfers through the tubes to the refrigerant. The refrigerant then becomes vaporized.

The evaporator tubes are continuously wetted by spraying refrigerant over the tube bundles. The refrigerant and vapor generated in this evaporative cooling process pass downward to the absorber, where the pressure is lowest in the system.

Absorber (low-pressure side)

This area of the machine is at a slightly lower pressure than that of the evaporator due to the absorption of vapor in the absorber. The pressure in this space, which is the rate of absorption, is controlled through the regulation of the absorbent solution concentration and temperature.

As the refrigerant vapor is absorbed, it is also condensed, releasing the heat of vaporization it acquired in the evaporator. This heat is rejected to the cooling tower water, which is circulated through the absorber tube bundle.

The absorber pump delivers large amounts of intermediate solution to the sprayers. A maximum surface area of solution is produced by spraying the solution over the tube bundle. This is necessary because absorption occurs only on the surface of the absorbent solution. This method also provides maximum heat transfer to the cooling tower water.

It is important to spray an intermediate solution rather than a concentrated solution for two reasons. First, a greater amount of solution is required to wet the tubes than is available from the concentrator. Second, if concentrated solution were sprayed directly onto the absorber tubes, it would be subjected to temperatures that could cause it to crystallize into a solidification of the lithium bromide.

Concentrator (high-pressure side)

To remove the refrigerant from the absorbent solution, diluted solution is pumped up to the concentrator. This diluted solution is then boiled, causing the refrigerant to leave the concentrator in the form of steam or hot water. When the steam or hot water leaves the concentrator, the solution left behind becomes more concentrated. The concentrated solution is then taken down and mixed again with dilute solution and re-enters the absorber as intermediate solution.

As the concentrated solution goes down and the dilute solution is pumped up, their pipes pass through a heat exchanger. Heat is transferred from the concentrator solution to the dilute solution, making the process more energy efficient.

Condenser (high-pressure side)

The refrigerant vapor that was produced in the concentrator migrates over to the cooler condenser where it clings to the tubes as it condenses. The cooling water

is the same that was circulated in a bundle through the absorber, so the water in the condenser bundle is at a higher temperature than that of the absorber. The condensed refrigerant is then passed through the orifice into the evaporator. At the low pressure in this location, some of the refrigerant flashes and cools the remainder of the refrigerant to evaporator temperature. This cooled refrigerant falls into the evaporator pan, ready to be sprayed over the tube bundle.

Pressure on the high side of the system is approximately 10 times higher than that of the low side, yet both are well below atmospheric pressure.

To get a better understanding of the absorption machine and visualize its actual appearance, see Figure C-8.

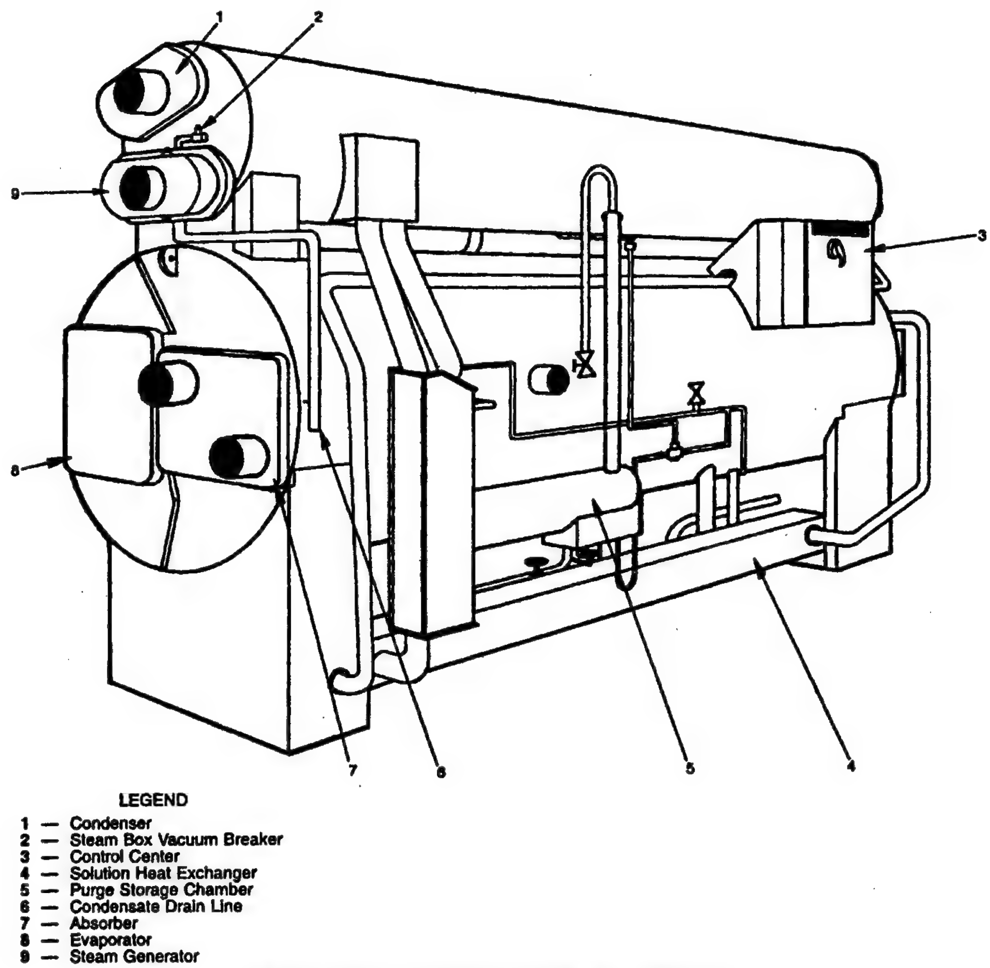


Figure C-8. Profile of Absorption Machine.

Source: Carrier Corporation, Syracuse, NY. Used with permission.

2 Applications of Chillers

Centrifugal

Heat recovery is one special application of centrifugal chillers. A separate closed condenser cooling water circuit is heated by the condensing refrigerant instead of rejecting all heat extracted from the chilled liquid to a cooling tower. This circuit is used for comfort heating, reheating, or preheating. Only one unit is designed for heat recovery in multiple packages.

Centrifugal chillers are capable of free cooling, which is cooling without the operation of the compressor. The chiller can operate as a thermal siphon when a supply of condenser water is available at a temperature below that of the needed chilled water temperature. Free cooling is limited to about 10 to 30 percent of the chiller design capacity.

Although the majority of centrifugal chilling units are for water-chilling applications, they can also be applied to brine cooling. Higher compressor speeds and more stages may be required due to the greater temperature lift.

Air-cooled centrifugal systems that are directly air-cooled eliminate the intermediate heat exchanger and condenser water pumps, which leads to lower power requirements. Condenser and refrigerant piping leaks have to be given special attention with this type of system. Also, this type of system should allow the condensing temperature to fall to around 70 °F during colder weather, which will lead to a decrease in compressor power consumption.

Reciprocating

Multiple reciprocating compressor units are widely used for the following reasons:

1. The number of capacity increments are greater, allowing extra standby capacity, lower power consumption, closer liquid temperature control, and less current in-rush during starting.

2. The potential for limited servicing or maintenance of some components while maintaining cooling is gained by multiple refrigerant circuits.

Reciprocating liquid chillers retain nearly full cooling capacity because pressure rise has only a slight influence on the volume flow rate of the compressor. These chillers are well suited for low-temperature refrigeration and air-cooled condenser applications.

The relationship between system demand vs. performance for a reciprocating liquid chiller is shown in Figure C-9 below. As cooling loads drop, compressor capacity also drops.

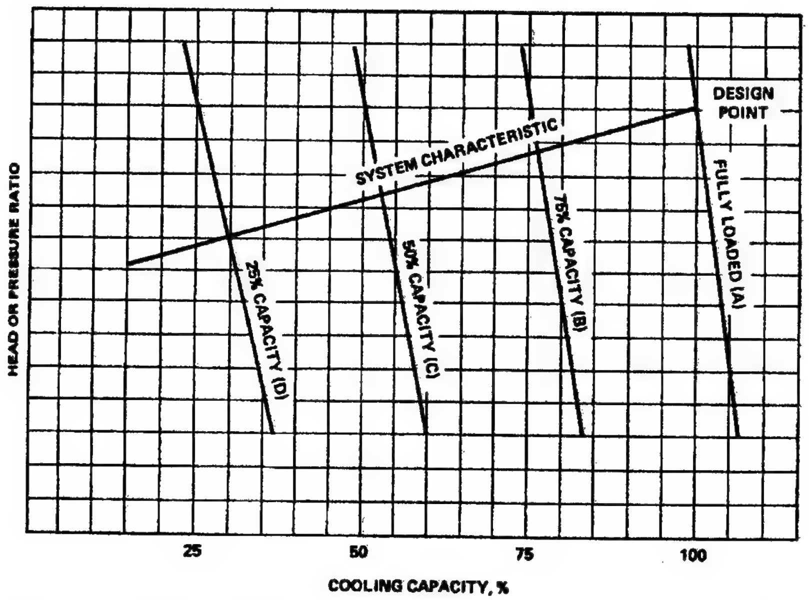


Figure C-9. Cooling Capacity vs. Performance for Reciprocating Chillers.
Reprinted with permission from the 1994 *ASHRAE Refrigeration Handbook.*

Screw

Screw compressors are being used for many applications due to its reasonable compressor cost, and no surge characteristics. Some of these areas are:

1. Heat-recovery installations
2. Air-Cooled
   a. Split packages with field-installed interconnecting piping
   b. Factory built rooftop packages
3. Low-temperature brine chillers for process cooling

Screw compressors are quieter and have only one-tenth the moving parts required in reciprocating compressors.

Absorption

The coefficient of performance (COP, see the efficiency section for further explanation) of absorption machines is lower than compression refrigeration systems. The COP for a large absorption machine is typically 0.65, while that of a large compressor driven water chiller may be 3.5 or higher. The energy use advantage of the compression refrigeration system is greatly reduced in actuality, and must be considered in application.

Moderate temperatures, supplied by flatplate solar collectors, can cool the water-lithium bromide absorption cycle at a high COP. The high COP is not as important in this case since there is no depletable fuel used.

Absorption-centrifugal combinations can be an efficient combination of refrigeration sources for air conditioning in some cases.

An absorption machine requires a larger cooling tower compared to that needed for a compression cycle machine. This is due to the larger quantity of heat that must be rejected from the absorber and condenser combined.

The choice of whether to use an absorption or vapor compression machine (or combination) for a specific application depends mainly on economics, which is a function of relative fuel costs.

3 Efficiency

Chiller efficiency (kW/ton) is a function of the percent of full load on the chiller and the refrigerant head. The refrigerant head is the refrigerant pressure difference between the condenser and evaporator, which is commonly represented by condenser water leaving temperature minus chilled water supply temperature (Figure C-10).

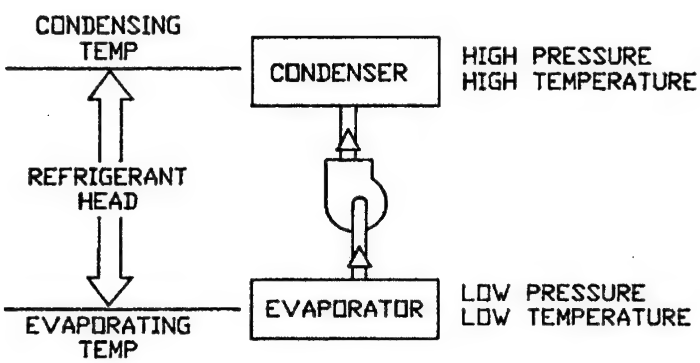


Figure C-10. Refrigerant Head.
Reprinted with permission from the 1995 *ASHRAE HVAC Applications Handbook.*

The chilled water supply temperature must be increased and/or the condenser water temperature decreased to reduce the refrigerant head. The gain is 1 to 2 percent energy savings for each degree Fahrenheit reduction in temperature.

Multiple chiller plants should be operated at the most efficient point on the part-load curve. The point at which a chiller should be added or dropped is shown in Figure C-11.

Coefficient of Performance (COP)

The COP is a factor that measures refrigeration requirements with power input. The COP is defined as:

$$COP = \frac{\text{Refrigeration Capacity}}{\text{Equivalent Power Input to Compressor}}$$

For a given refrigeration requirement, a greater power is necessary for a lower COP of a refrigeration unit.

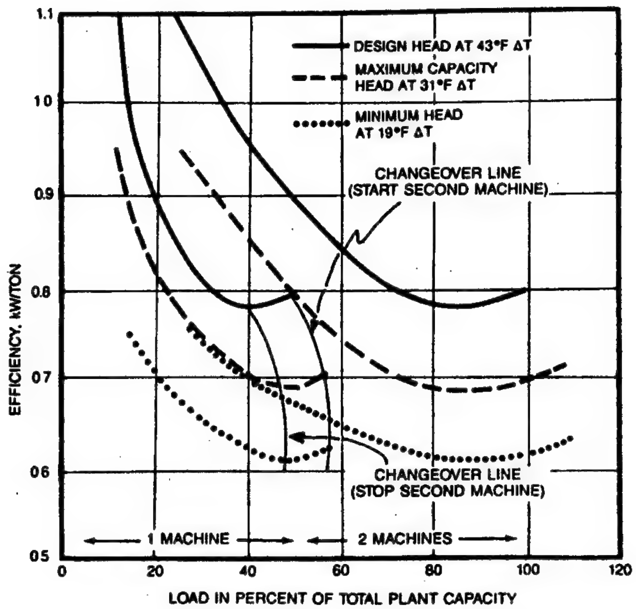


Figure C-11. Multiple Chiller Operation Changeover Point—Two Equal Sized Chillers.
Reprinted with permission from the 1995 *ASHRAE HVAC Applications Handbook.*

Example 1

A packaged chiller is known to have a 28.1 ton capacity. The chiller requires 25.1 kW to operate at this capacity.

$$\text{capacity} = 28.1 \text{ tons} \times 12{,}000 \frac{\text{Btu/hr}}{\text{ton}} = 337{,}200 \text{ Btu/hr}^{*}$$

$$\text{power input} = 25.1 \text{ KW} \times 3{,}410 \frac{\text{Btu/hr}}{\text{KW}} = 85{,}590 \text{ Btu/hr}^{**}$$

$$\text{COP} = \frac{337{,}200 \text{ Btu / hr}}{85{,}590 \text{ Btu / hr}} = 3.94$$

---

* Enthalpy of fusion for water = 144 Btu/lb (required energy)

$$1 \text{ ton} = 1 \text{ ton} \times \frac{2000 \text{ lb}}{\text{ton}} \times \frac{144 \text{ Btu}}{\text{lb}} \times \frac{1}{24 \text{ hr}} = 12{,}000 \text{ Btu/hr}$$

* KW = power; conversion: 1 KW = 3,410 Btu/hr

4 Chiller Components

Types of Compressors

Compressors may be classified as open, hermetic, or semihermetic.

Open Compressors

Open compressors (Figure C-12) require an external driver. Electric motors are most common, but steam or internal combustion engines can also be used. The external driver is attached to the compressor crank shaft either directly with a coupling or belt driven to operate at a specific speed. The external drive and compressor are in two separate housings. Open compressors are generally more expensive than hermetic and semihermetic compressors because of these separate housings.

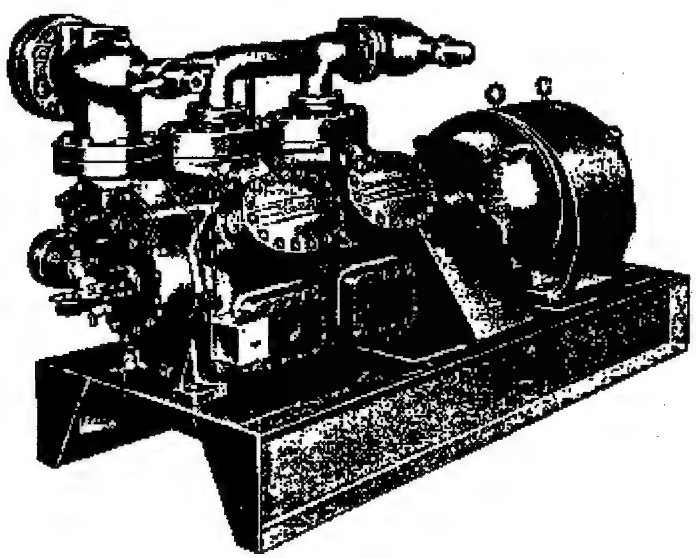


Figure C-12. Open Compressor Unit.

Source: The Trane Company, La Crosse, WI. Used with permission.

Hermetic Compressors

Hermetic compressors (Figure C-13) have the motor and compressor enclosed in one housing. The compressor and motor share a common shaft and bearings.

Motors are usually suction gas cooled as the rotor is mounted on the compressor crankshaft. Generally the horsepower of the motor is matched to the compressor and refrigerant. The only external connections required are wiring and piping. Factory repairs are necessary as the compressor is hermetically sealed (welded).

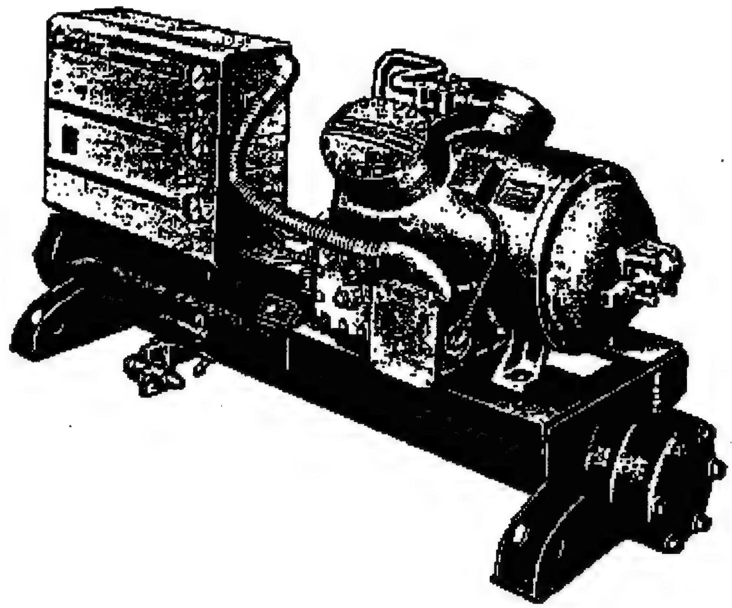


Figure C-13. Hermetic Compressor Unit.

Source: The Trane Company, La Crosse, WI. Used with permission.

Semihermetic Compressors

The difference between hermetic and semihermetic compressors is that semihermetic compressors may be serviced in the field because they are manufactured with bolted means of access. With proper controls, all forms of compressors can be used in all applications. In some cases, the size of the compressor will determine what type of compressor is used.

Condenser

The condenser is the component of the mechanical refrigeration cycle that rejects and removes heat. This includes heat from the evaporator plus the heat equivalent of the work of compression. The three basic types of condensers are:

1. Water-cooled
2. Air-cooled
3. Evaporative.

Water-Cooled

Most water-cooled condensers in use today can be classified into two categories: shell-and-coil or shell-and-tube.

Shell-and-coil condensers circulate cooling water through one or more continuous or assembled coils. Refrigerant vapor is condensed outside the tubes contained within the steel shell. Horizontal or vertical shell arrangements are available in sizes ranging anywhere up to 20 tons. This type of condenser is small, compact, and efficient and may be cleaned by chemical means.

Shell-and-tube condensers (Figure C-14) circulate cooling water through tubes in a single- or multi-pass circuit. Water flows within the tubes, and refrigerant vapor fills the space between the shell and the tubes.

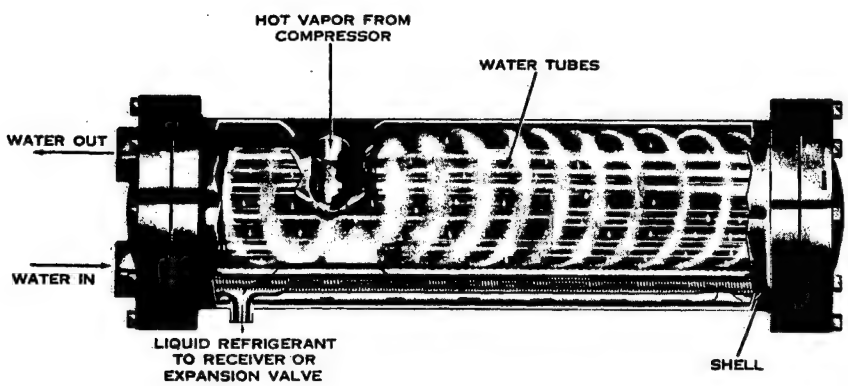


Figure C-14. Shell-and-tube Condenser.
Enrivonmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

A pickup tube or sump is usually provided at the bottom of the shell to collect the condensed refrigerant. Horizontal arrangements are typical for shell-and-tube condensers. Sizes range anywhere up to 10,000 tons, and tubes may be cleaned mechanically. Cooling towers are commonly used with both types.

Air-Cooled

Air-cooled condensers (Figure C-15) operate by circulating refrigerant through a coil, with air flowing across the outside of the tubing. Coils are commonly made of copper, aluminum, or steel tubes with diameters ranging from 0.25 to 0.75 in.

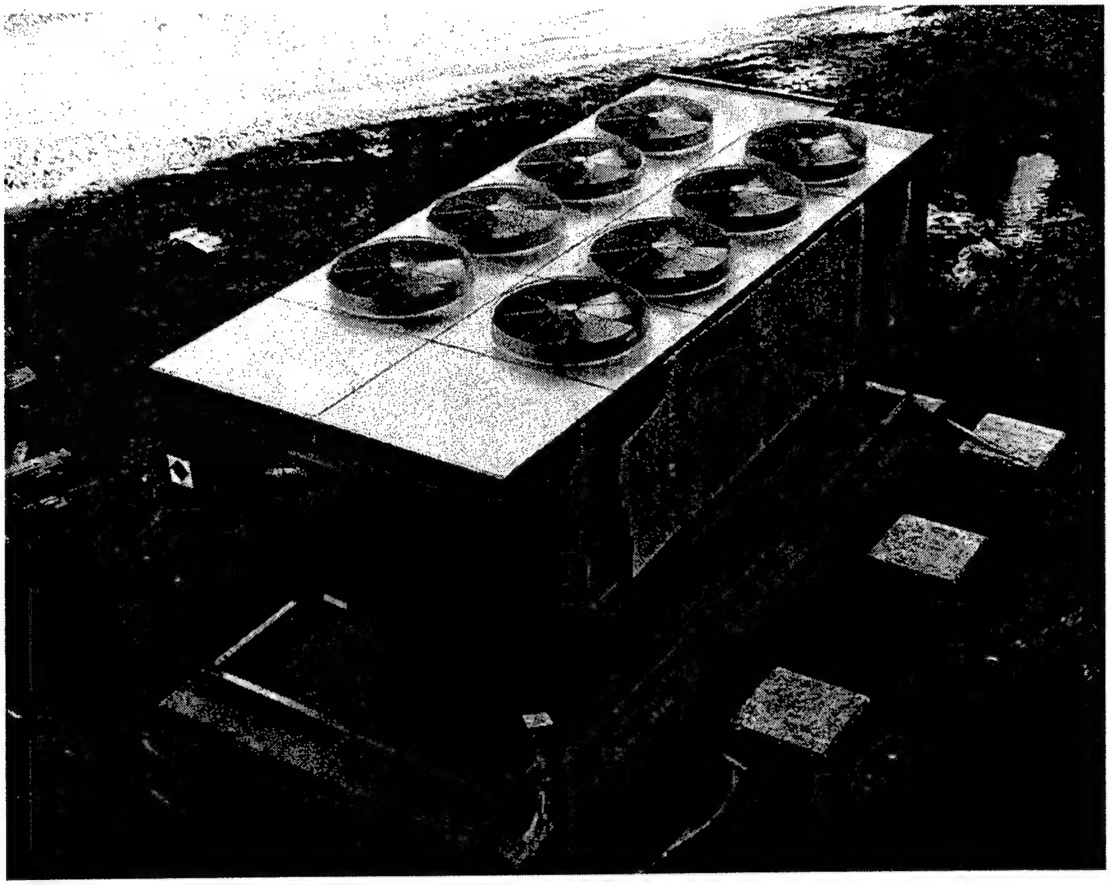


Figure C-15. Air-cooled Condenser.

Air-cooled condensers produce air motion by natural convection or by the use of a fan. Natural convection condensers are limited to smaller capacities. Indoor or outdoor locations can be used. In outdoor applications, a clean area should be selected that positions the condenser towards prevailing winds. Compressor capacity can be increased by adding a liquid subcooling circuit.

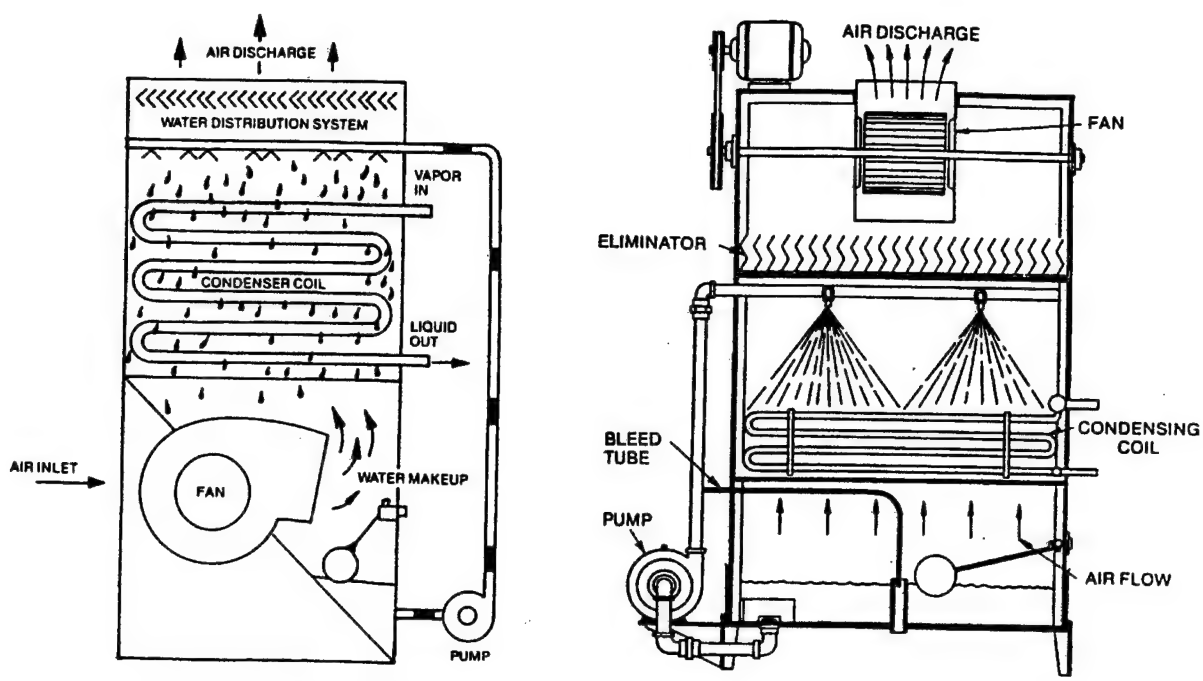
Forced-air condensers are typically used when the condenser is remote from the compressor. Remote condensers can be located indoors or outdoors. The greater the distance between the condenser and compressor, the greater the first cost and operating cost.

Maintenance for air-cooled condensers is simple. They do not have to be started up in the spring or winterized in the fall. The only connections required are refrigerant and power, and they are easy to install.

Evaporative

Refrigerant vapor from the compressor enters the top of a coil and condenses to a liquid as it flows through the coil. Water is sprayed down over the refrigerant coil with the spray water falling into a water tank to be picked up by a pump and returned to the spray nozzles. Air is simultaneously directed over the coil, causing a small portion of the recirculated water to evaporate. This evaporation removes heat from the coil, cooling and condensing the vapor. The liquid refrigerant then drains to a receiver.

Coils are commonly made of steel or copper tubing. Evaporative condensers (Figure C-16) can be arranged horizontally or vertically. Outdoor installation is commonly used, and freeze-up problems must be considered.



(a) Blow-through Type (b) Draw-through Type

Figure C-16. Evaporative Condensers.
Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

Flow Components

Figure C-17 shows a chiller's flow components, which are described below.

Flow Control Device

This restricting device regulates the refrigerant flow according to the load and causes the pressure drop of the refrigerant. Some common types used are:

1. Thermostatic expansion valve
2. Capillary tube
3. Low-side float valve.

Thermostatic expansion valve. The thermostatic expansion valve (Figure C-18) is used in direct-expansion systems. Refrigerant flow is regulated automatically by the valve reaction to the pressure variations in the remote bulb being transmitted through the tube to the thermal valve. When the bulb senses a temperature below the control point, the thermal valve is throttled.

Capillary tube. The capillary tube is a small-diameter tube that connects the outlet of the condenser to the inlet of the evaporator, resulting in the required pressure drop. It is used with direct-expansion systems.

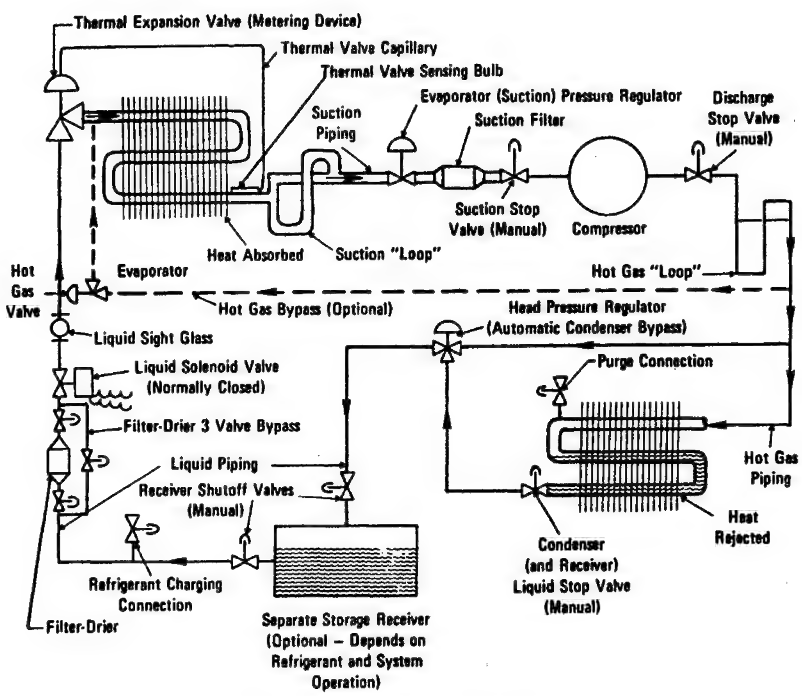


Figure C-17. Flow Components.

Enrivonmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

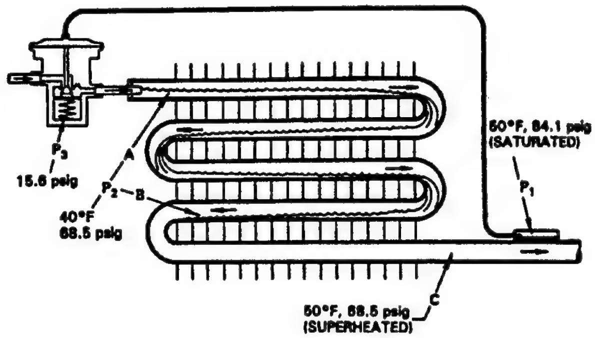


Figure C-18. Thermostatic Expansion Valve.

Reprinted with permission from the 1998 *ASHRAE Refrigeration Handbook.*

Low-side float valve. Low-side float valves are used with flooded systems. If flow is not adequate and too much liquid accumulates, the float rises and a connecting linkage opens the valve, allowing more flow.

Suction Piping

This piping returns refrigerant vapor to the compressor. It is the most important piping in the system because it has to be large enough to maintain minimum friction to prevent reduced compressor capacity, but also must be small enough to generate enough gas velocity to return the system oil to the compressor.

Evaporator Pressure Regulator

This automatic pressure regulator maintains constant pressure in the evaporator. It helps prevent freezing of water and limits minimum relative humidity at light loads in any refrigeration system.

Suction Line Filter

This filter is generally used with steel piping systems to remove rust. Pressure drops in the line are associated with this device.

Discharge Stop Valve

The manual service valve at the leaving connection of the compressor.

Suction Stop Valve

Manual service valve at the inlet side of the compressor.

Receiver

An auxiliary storage space to store refrigerant when system is shut down, or needs to be opened for servicing.

Refrigerant Charging Connection

Manual valve that introduces refrigerant into the system.

Filter-Drier

This device used to strain and remove moisture. It is normally used with a three valve bypass to allow removal when in operation. This device does not need to be used on smaller systems but is recommended for all systems.

Liquid Solenoid Valve

Electrically operated control valve located in the liquid piping that can stop refrigerant flow.

Liquid Sight Glass

Glass-ported fitting in the liquid refrigerant line, located immediately ahead of the expansion valve, that provides a means for viewing the liquid flow.

Hot Gas Bypass and Valve

The piping and manual (but more often automatic) valve used to introduce compressor discharge gas directly into the evaporator.

Relief Devices

Relief valves or rupture discs are used to relieve excess pressure and are commonly piped to the outdoors. These devices are required by code.

Cooler (Evaporator)

Refrigerant is vaporized in the cooler. Refrigerant evaporates inside the tubes of a direct expansion cooler. These coolers are usually used with positive-displace-

ment compressors to cool water or brine. In a flooded cooler, the refrigerant vaporizes on the outside of tubes, which are submerged in liquid refrigerant within a closed shell. Flooded coolers are usually used with screw or centrifugal compressors. Direct-expansion coolers do not require liquid storage, whereas flooded coolers maintain a liquid pool of refrigerant. The four basic types of coolers are: (1) Shell-and-Tube Cooler (DX, Flooded), (2) Baudelot Cooler (DX, Flooded), (3) Shell-and-Coil Cooler, and (4) Direct Expansion Cooling Coil.

Shell-and-Tube Cooler

In a direct-expansion shell-and-tube cooler (Figure C-19), refrigerant circulates through the tubes in a single or multi-pass circuit. Fluid baffles (plates used to control liquid flow) on the outside of the tubes channel fluid flow and, in turn, increase the velocity of the fluid. Refrigerant distribution is critical, as tubes that are fed more refrigerant than others tend to bleed into the suction line.

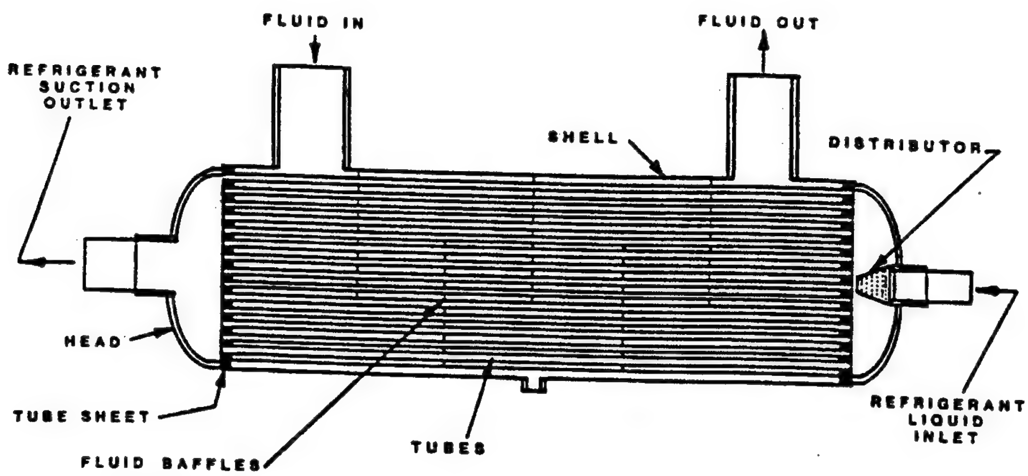


Figure C-19. Direct-Expansion Shell-and-Tube Cooler.
Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

Flooded shell-and-tube coolers (Figure C-20) vaporize refrigerant on the outside of tubes. Fluid flows through the tubes, which are submerged in refrigerant, and all are contained in a closed shell. Refrigerant is usually fed into the bottom of the shell by a distributor that equally distributes it under the tubes. Warm fluid in the tubes heats the refrigerant, causing it to boil.

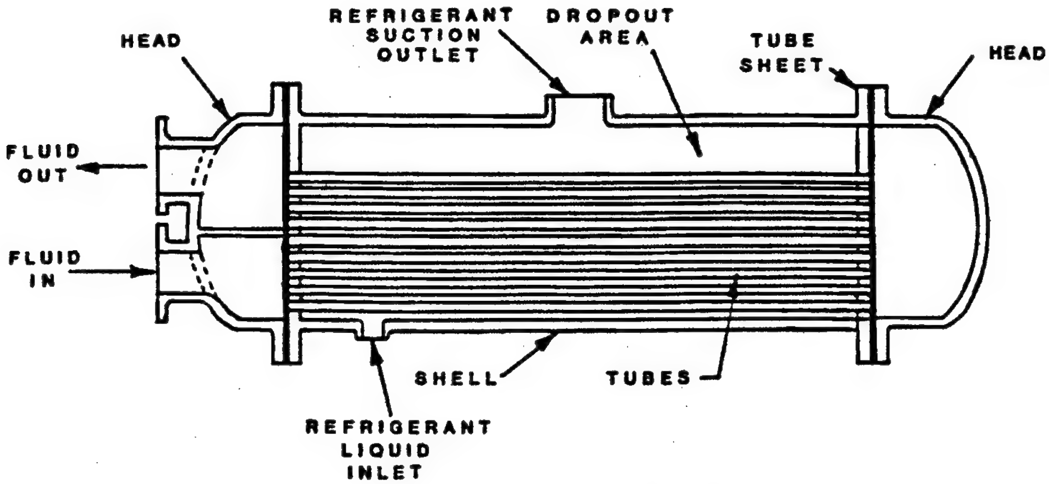


Figure C-20. Flooded Shell-and-Tube Cooler.

Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

Baudelot Cooler

Baudelot coolers (Figure C-21) may be set up for flooded or direct-expansion operation. The fluid to be cooled is distributed over the heat exchanger and then flows by gravity to a collection plate below. Vertical plates or horizontal tubes are used in the heat exchanger to allow easy cleaning.

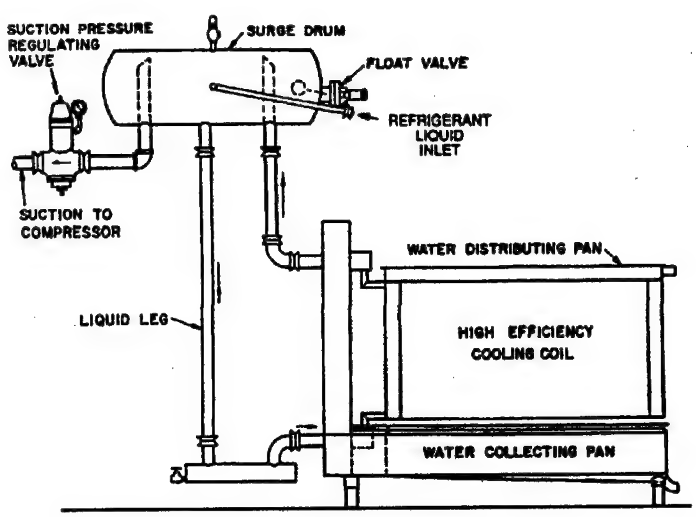


Figure C-21. Flooded Baudelot Cooler.

Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

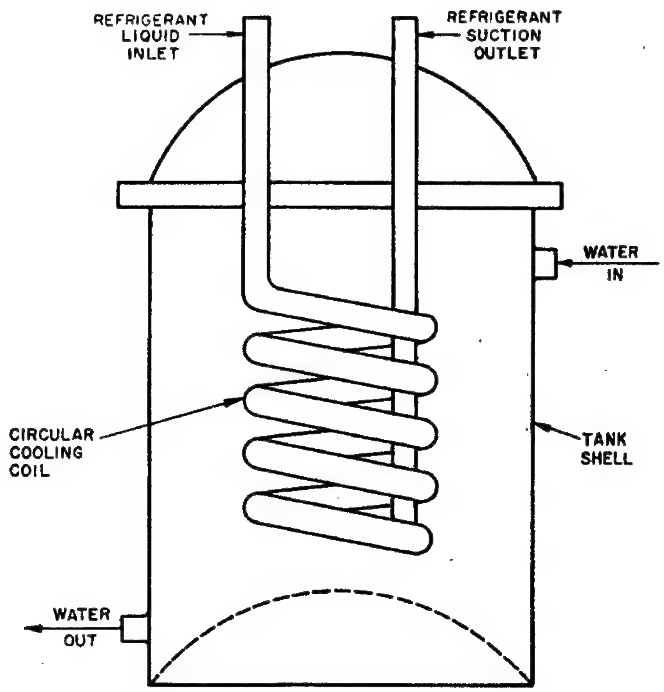


Figure C-22. Shell-and-Coil Cooler.

Reprinted with permission from the 1996 *ASHRAE Systems and Equipment Handbook.*

Shell-and-Coil Cooler

Shell-and-coil coolers (Figure C-22) consist of a coiled tube in which the refrigerant flows and a tank that contains the fluid to be cooled. The coiled tube can be located either outside or inside the tank. In some cases, the tank can be opened for cleaning.

Direct Expansion Cooling Coil

Coil equipment (Figure C-23) used for cooling an airstream under forced convection may consist of a single coil section or a number of individual coil sections built up into banks and assembled in the field. Coils are used to cool air whereas coolers cool water or brine.

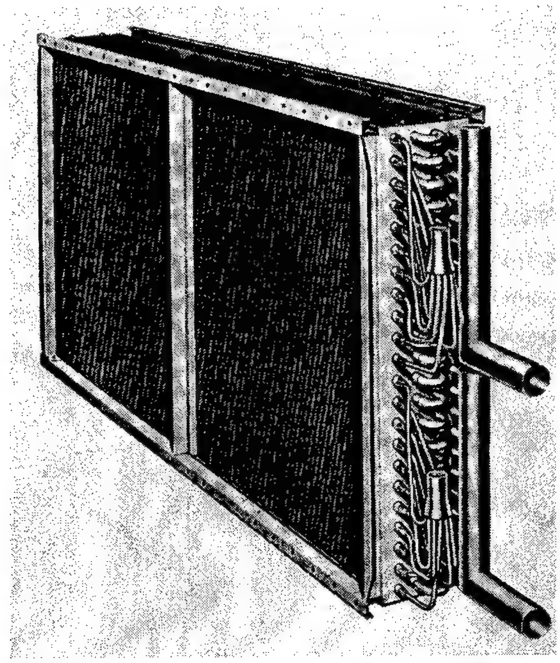


Figure C-23. Direct Expansion Cooling Coil.

Enrivonmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

Cooling Towers

The heat generated by the operation of HVAC systems must be dissipated. Water is commonly used as a heat transfer medium to remove heat from refrigerant condensers or heat exchangers. In the past, this was accomplished by bringing large amounts of water from a natural or infinite source, heating it by thermodynamic heat transfer processes, and then discarding it back to the natural environment. Today the cost of water from utility services, and the cost of disposing it, is very high and unreasonable for cooling a large system.

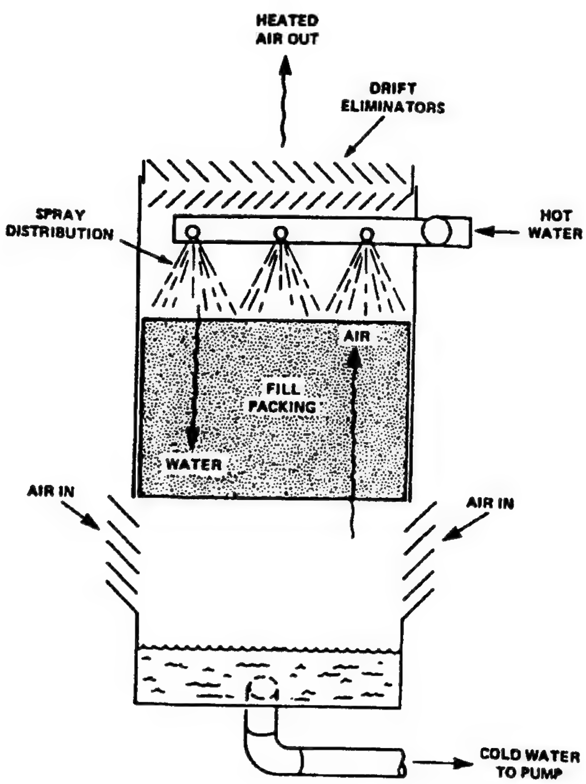
Cooling towers overcame several problems that became apparent with the old methods of cooing a system. The water consumption rate of a cooling tower system is only about 5 percent of that of a once-through system, making it the least expensive system to operate with purchased water supplies (ASHRAE 1988). Also, cooling towers can cool water to within 5 to 10 °F of the ambient wet-bulb temperature or about 35 °F lower than air-cooled systems of reasonable size.

Types of Cooling Towers

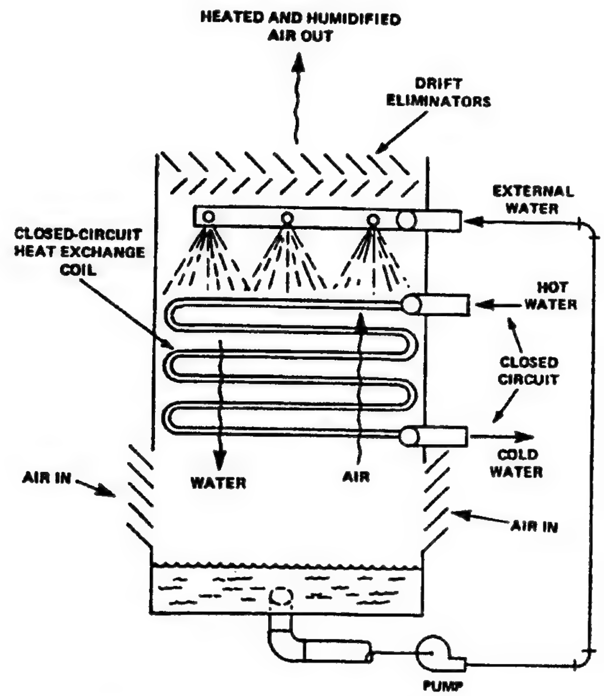
Figure C-24 shows the two basic types of evaporative cooling:

1. An external circuit which exposes water to the atmosphere as it cascades over the tubes of a coil bundle (direct-contact evaporative cooling tower).
2. An internal circuit in which fluid to be cooled is circulated inside the tubes of the coil bundle (indirect-contact evaporative cooling tower).

The internal fluid circuit is advantageous when the fluid inside the tubes is used to cool fluids other than water and to prevent contamination of the primary cooling circuit with airborne dirt and impurities. Heat transfers through the pipe walls from the internal fluid to the walls of the pipe and is finally absorbed into the external water circuit, which is cooled evaporatively.



(a) Direct-Contact Evaporative Cooling Tower



(b) Indirect-Contact Evaporative Cooling Tower

Figure C-24. Cooling Tower Types.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

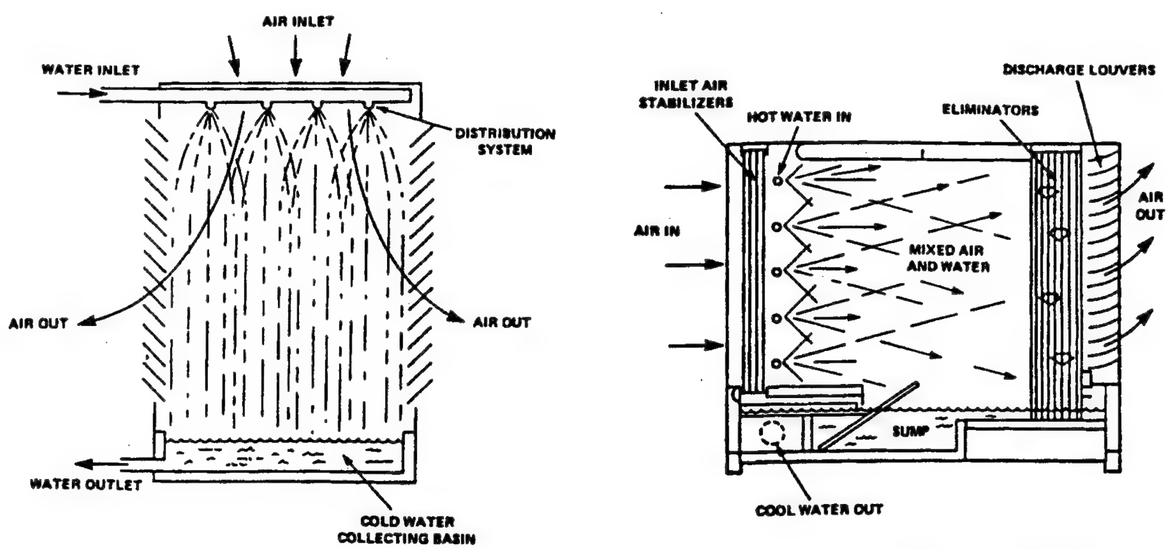
Spray-filled towers are a common method of exposing water to air in direct-contact devices where water is exposed to air without use of a heat transfer medium. There are several ways to increase contact surfaces as well as time of exposure of the water. One method is to use a heat transfer medium or fill by installing it below the water distribution system in the path of the air.

Two types of fill are splash-type and film-type. The splash-type fill maximizes contact area and time by forcing water to cascade through successive elevations of splash bars arranged in staggered rows. Film-type fills cause the same effect by having the water flow in a thin layer over closely spaced sheets (usually PVC) that are arranged vertically.

For thermal performance levels usually found in air conditioning and refrigeration, the tower with film-type fill is usually more compact. Splash-type fill is less sensitive to initial air and water distribution and is usually the fill of choice for water qualities that are conducive to plugging.

Direct Contact Cooling Towers

Nonmechanical draft towers. This type of cooling tower is aspirated by sprays or density differential and does not contain a fill or use of a mechanical device such as a fan. The aspirating effect of the water spray, either vertically or horizontally, induces airflow through the tower in a parallel flow pattern as depicted below in Figure C-25.



(a) Vertical Spray Tower (b) Horizontal Spray Tower

Figure C-25. Nonmechanical Draft Towers.

Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

Mechanical draft towers. When the fan in a cooling tower is on the inlet air side, it is called forced-draft. When the fan is on the exit air side, it is known as induced draft. Depending on the external pressure needs and sound level acceptance, a centrifugal or axial fan is chosen.

In mechanical draft towers, water flow is downward and airflow is either upward (counterflow heat transfer) or horizontal (crossflow heat transfer). Air may enter one side or two sides of the tower.

Towers are classified as either factory-assembled or field-erected. Figure C-26 shows the various types of mechanical draft cooling towers.

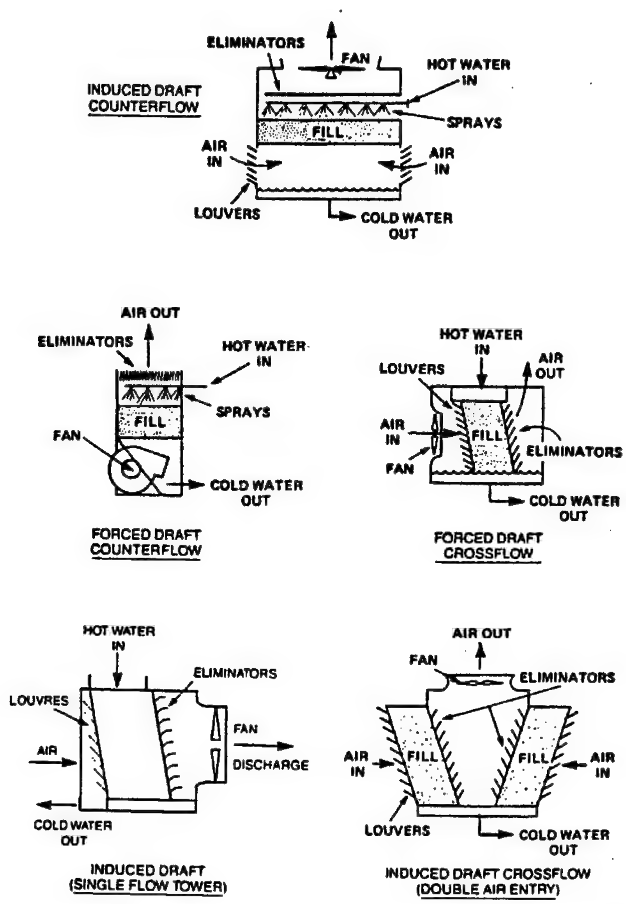


Figure C-26. Conventional Mechanical Draft Cooling Towers.
Reprinted with permission from the 1992 *ASHRAE HVAC Systems and Equipment Handbook.*

Indirect Contact Cooling Towers

Closed circuit fluid coolers (mechanical draft). Counterflow and crossflow types are used in forced and induced fan arrangements as in Figure C-24. The tubular heat exchangers are typically serpentine bundles, usually arranged for free gravity internal drainage. Pumps are used to transport water from the lower collection basin to the upper distribution basin or sprays. The internal coils are predominantly fabricated of galvanized steel or copper. Closed circuit fluid coolers that are similar to evaporative condensers are increasingly used on heat pump systems and screw compressors.

Coil shed towers (mechanical draft). In coil shed towers, both crossflow and counterflow types are available with either induced or forced fan arrangements. These towers usually have isolated coil sections that are nonventilated, and are located beneath a conventional cooling tower.

Redistribution water pans located at the tower's base feed cooled water by gravity flow to the tubular heat exchange bundle (coils). These units are typically arranged as field-erected, multifan cell towers and are used primarily in the process cooling industry.

Selection Considerations

Choosing the right water-cooling equipment for a specific application requires some insight into the cooling duty, economics, required service, and environmental conditions. Although each of these considerations are integrated in some respects, they should be evaluated separately.

Also influencing the selection of equipment will be other physical parameters such as size, height, length, width, plan area, volume of airflow, fan and pump energy consumption, materials of construction, water quality, and availability.

The optimum choice is generally made after an economic evaluation is completed. Some initial cost considerations might include:

1. Erected cost of equipment
2. Cost of interphase with other equipment or subsystems that include:
   a. Basin grillage* and value of space occupied.
   b. Cost of pumps.
   c. Electrical wiring to pump and fan motors.
   d. Electrical controls and switch gear.
   e. Cost of piping to and from tower.
   f. Tower basin, sump screens, overflow piping, and makeup lines (if not provided by the manufacturer).
   g. Shut off and control valves (when not furnished by the manufacturer).
   h. Walkways, ladders, etc. to provide access to the tower.

* Basin grillage is used to reduce turbulence in the basin.

Ownership and Maintenance Costs

Things to consider in the long term of ownership are system energy costs (fans, pumps, etc.) on the basis of operating hours per year, demand charges, expected equipment life, maintenance and repair costs, and initial costs.

In addition to the long-term costs, some elements are required by code. Note the following:

- Safety features and safety codes
- Conformity to building codes
- General design and rigidity of structures
- Relative effect of corrosion, scale, or deterioration on service life
- Availability of spare parts
- Reliability of manufacturers.

Refrigerants

Refrigerants are the working fluids in refrigeration systems. They absorb heat from the system by evaporating and then dispose of the heat by condensing from a vapor back into a liquid. This process can occur in both the mechanical compression system and the absorption system.

Choosing a refrigerant for a particular application depends on several properties:

- *Flammability:* Ability to burn or support combustion.
- *Toxicity:* Poison content.
- *Specific Volume:* Volume of fluid per unit mass.
- *Normal Boiling Point:* Temperature at which the vapor pressure of a fluid is one standard atmosphere (14.696 psi). This measurement can be a direct indicator of the temperature level at which a refrigerant can be used.
- *Viscosity:* Internal frictional resistance exhibited by a fluid in resisting a force that tends to cause the liquid to flow.

Compressors can use refrigerants with properties besides those listed above. The choice of refrigerant will depend on the specific characteristics of that system and the system's cooling requirements.

Centrifugal

Centrifugal compressors work well in handling relatively high flow rates of suction vapor. Volumetric flow of suction vapor increases with lower suction temperatures and higher capacities, and higher pressure refrigerants like R-12 and R-22 work well.

The weight of refrigerant piping as well as the physical size and other components of the refrigeration system are reduced by the use of higher pressure refrigerants. R-113, R-11, R-114, R-12, R-500, or R-22 are commonly used due to this reason.

Reciprocating

The compressor size can be reduced and the total chiller price is low if the compressor displacement required for a given capacity is minimized. R-22 is commonly used for this reason. R-717 has the same advantage, but it is incompatible with copper and has an unpleasant odor.

Low temperature applications usually use R-22, R-502, and R-717. High condensing temperature applications, such as heat recovery units or heat pumps, can use R-12 or R-500 due to lower discharge temperatures and condensing pressures.

Discharge gas temperatures do not increase rapidly with the compression ratio and make R-12 and R-502 suitable for low evaporating and high condensing temperature applications.

Screw

R-500 and R-12 are usually used in heat-recovery installations where temperatures of the heat-transfer media range up to 158 °F. R-22 and R-717 are commonly used since screw compressor sizes are relatively small.

Alternative Refrigerants

As the production of chlorofluorocarbons (CFCs) is reducing, existing chillers need to be retrofitted with alternative refrigerants. The Clean Air Act of 1990 established legislation on the elimination of the use of CFCs. The Act calls for incrementally reducing and then terminating all production of CFCs, halons, and

other ozone-destructive chemicals by the year 2000 in developed nations, and by 2010 for developing nations. This accelerated phaseout is expected to cause shortages of CFCs before the end of the decade. Also, an increasing tax rate went into effect in 1990 to discourage their use.

Substitute refrigerants have been developed to replace those with CFCs. One of these substitutes is HFC-134a (R-134a), which contains no chlorine, and has no effects on the ozone layer. Other substitutes are HCFC-123 (R-123) and HCFC-22 (R-22), which do contain chlorine, but break down in the atmosphere much faster than CFCs. These substitutes, while effective in reducing the amount of CFCs in the atmosphere, operate less efficiently than current refrigerants. This means that more fossil fuels are burned to make up for the lower efficiency. This increased fossil fuel consumption limits the effectiveness of these substitutes as viable alternatives.

These substitute refrigerants are limited in use as well by the Clean Air Act. The Act allows manufacturers to sell (until 2020) new refrigeration equipment that uses HCFCs. After 2020, only service on existing equipment will be allowed. The Act also freezes production levels of HCFCs beginning in 2015 and bans all production in 2030. The final solution to this problem has yet to be determined, but it will have a large effect on all refrigeration systems.

Retrofitting equipment currently using R-11 may involve replacing seals, gaskets, bushings, motor insulation, compressor motor, or diaphragms to use R-123. Because of its pressure rating, R-123 is the only alternative for R-11 at this time. R-123 has a higher specific volume and lower acoustic velocity than R-11. An R-123 compressor will have to circulate 10 to 15 percent more inlet cubic feet per minute and generate about 7 percent more lift or head given the same capacity requirements. Compressor capacity losses range from 0 to 18 percent.

Medium pressure chillers, which now use R-12 or R-500, will be retrofitted with R-134a. Changing to this refrigerant will typically require a gear drive change. The combination of higher acoustic velocity and higher suction specific volume, (characteristics of R-134a) results in a higher required compressor lift or head.

5 Design

Sizing and Specifying Chillers

Costs

In times of energy awareness and cost-effective spending, it is important that the cooling load size be calculated accurately. If the total required chiller capacity is not known accurately, the chiller owning cost may be unnecessarily high, as cooling load size is one of the largest factors in the cost of owning a chiller.

Oversized equipment can cause problems. One problem is the surging or frequent on/off cycling of centrifugal machines at low loads. When methods of estimating loads are used, the practice of adding a 10 to 20 percent safety factor is unnecessary because of the availability of accurate methods. Adding the safety factor also proportionately increases the cost of purchase, installation, and poor efficiency from wasted power.

The undersizing of equipment should be considered unacceptable in practice, yet the problem of undersizing is not of such serious consequences as oversizing. The consequence of a small underestimation is the increase in chilled liquid temperatures for a few design load days of the year.

The total cost of ownership is the primary criterion for equipment selection. Such criterion is comprised of the following:

- *Purchase Price:* Each machine type and manufacturer's model should include all the necessary auxiliaries such as starters and vibration mounts. If they are not included, their price should be added to the base price.

- *Installation Cost:* Factory-packaged machines are less expensive to install and usually considerably more compact, resulting in space savings. The cost of field assembly of field erected chillers should also be evaluated.

- *Energy Cost:* If using an estimated load schedule and part-load power consumption curves furnished by the manufacturer, a year's energy cost should be calculated.

- *Maintenance Cost:* Each bidder could be asked to quote on a maintenance contract on a competitive basis.

- *Insurance and Taxes:* Hermetic units often require higher insurance premiums.* Purchase price is the most used criterion, yet the cost of energy is causing the operating cost to be more deeply scrutinized in certain sectors. Package arrangements and accessories which offer increased operating economy are increasing in use.

Methods of Selection

The following is used as a guide for determining the types of liquid chillers generally used for air conditioning:

up to 25 tons (88 kW)	Reciprocating
25 to 80 tons (88 to 280 kW)	Reciprocating or Screw
80 to 200 tons (280 to 700 kW)	Reciprocating, Screw, or Centrifugal
200 to 800 tons (700 to 2800 kW)	Screw or Centrifugal
above 800 tons (2800 kW)	Centrifugal

For air-cooled condenser duty, brine chilling, or other high head applications from 80 to 200 tons (280 to 700 kW), reciprocating and screw liquid chillers are more frequently installed than centrifugal.

Reciprocating. Two types of ratings are published:

- *Packaged Liquid Chiller:* This lists values of capacity and power consumption for many combinations of leaving condenser water and chilled water temperatures.

* When the unit breaks down, the entire unit must be replaced.

- *Capacity and Power Consumption at Varying Temperature:* Specific equipment such as remote condenser, evaporative, water-cooled, or air-cooled chillers are selected to achieve these requirements.

With all liquid chilling systems, condensing temperature increase means greater power consumption. So, the smallest packaged chiller, with the lowest ratio of input to cooling capacity, can be used for the following: low condenser water temperature, relatively large remote air-cooled condenser, or when leaving chilled water temperature is high. Just because the liquid chiller cost is minimized does not mean the cost of the total system will be low. For example, an increase in cooling tower or fan coil cost will reduce or offset the benefits of reduced compression ratio.

Centrifugal. The details specified for centrifugal systems include the number of passes in each of the heat exchangers and may include changes in rated motor kilowatt capacity of turbine size, code indication for driving gear ratio, and code indication of impeller diameters.

The maximum number of condenser and cooler water passes should be used without producing excessive water pressure drop. The greater the number of waterside passes, the less the power consumption.

Noise and vibration control are another consideration in selecting equipment (see acoustics in the **Installation** section).

Screw. Screw chiller ratings are presented similarly to those of the centrifugal-chiller ratings. Tabular values of capacity and power consumption at various chilled water and condenser water temperatures are given.

In addition, ratings are given for packages minus the condenser, listing capacity, and power vs. chilled water temperature and condensing temperature. Ratings for compressors alone are also common.

Codes

Refrigerants

The 1988 edition of the Uniform Mechanical Code (UMC) requires that refrigerants be classified in the following two groups.

Group 1: Refrigerants are noncorrosive, nonflammable, nontoxic, nonexplosive, and can be used in HVAC systems.

R-11	R-21	R-115
R-12	R-22	R-C318
R-13	R-30	R-500
R-13B1	R-113	R-502
R-14	R-114	R-744

Group 2: Refrigerants can be used in some process installations but are generally not used in HVAC systems because they are toxic.

R-40

R-611

R-717 (ammonia)

R-764

The most common refrigerants used in HVAC systems are R-11, R-12, R-22, R-113, R-114, and R-500.

Condensing units or combinations of refrigerant-interconnected condensing units that contain a Group 1 refrigerant and totals a 100 horsepower rating or more shall be enclosed in a refrigeration machinery room (exceptions to this are allowed; consult UMC).

A refrigerating system containing a Group 2 refrigerant will not be located within a building unless all refrigerant-containing portions of the system are enclosed in a refrigeration machinery room. If installed outside, it shall be located 20 ft or farther from any window, ventilating-air inlet, or exit door in a building.

Machinery Rooms

Code required machinery rooms will be built of 1-hour (or greater) fire-resistive construction. Doors will open in the direction of egress, and comply with the Uniform Building Code (UBC). Openings that would permit the passage of escaping refrigerant to other parts of the building are not allowed. Machinery rooms will be 50 sq ft in area or larger.

All moving machinery contained in these rooms will have at least a 2 ft 6 in. wide by 7 ft high unobstructed working space extending around two adjacent sides of the equipment.

At least one exit door 3 ft by 6 ft 8 in. or larger shall be used with equipment containing Group 1 refrigerants. At least two exit doors, located at least one-fifth the perimeter of the room apart, shall be used with equipment containing Group 2 refrigerants. These doors will be at least 3 ft wide by 6 ft 8 in. in height.

Absorption systems containing a Group 2 refrigerant will be installed in a refrigeration machinery room.

Section 1507 of the UMC states that: "There shall be no direct opening between a refrigeration machinery room containing a Group 2 refrigerant, and a room or space in which there is an open flame, spark-producing device, or heating surface hotter than 800 °F."

Ventilation requirements for machinery rooms and other rooms containing portions of a condensing unit should be followed. Restrictions concerning the location of electrical equipment within a machinery room should also be followed.

Clearances and Supports

The 1988 edition of the UMC requires that: "A compressor or portion of a condensing unit supported from the ground shall rest on a concrete or other approved base extending not less than 3 inches above the adjoining ground level." This requirement also pertains to absorption systems. In addition to this, aboveground platforms used for evaporative coolers will be 6 in. above the adjoining ground level.

UMC also required that a 2 ft or greater unobstructed access opening and passageway be provided and maintained to a compressor.

Absorption systems containing a Group 2 refrigerant weighing more than 20 lb will be located 20 ft or more from any window, door, or ventilating air inlet to a building. Absorption systems containing a Group 2 refrigerant will not be located in any building unless installed within a mechanical room.

Equipment

Piping and tubing shall have points of support every 15 ft or less. A securely fastened permanent support shall be provided within 6 ft following the first bend in tubing from the compressor, and each other bend or angle shall have these supports within 2 ft. Piping crossing an open passageway shall be a minimum of 7-1/2 ft above the floor unless it is against the ceiling in the space. Piping and tubing should be checked with UMC pertaining to size, use, refrigerant, and material.

A stop valve will be installed in refrigerant piping at the outlet and inlet of every positive-displacement type compressor; at each refrigerant outlet from a receiver; and at each refrigerant inlet of a pressure vessel containing liquid refrigerant and in excess of 3 cu ft in internal gross volume. Stop valves made of copper tubing 3/4 in. or less outside diameter will be supported independent of the tubing or piping connected to the valve.

A pressure-limiting device will be installed on a positive displacement refrigerant compressor that is a portion of: a system containing Group 2 refrigerant, an air-cooled system containing Group 1 refrigerant that is of 10 hp or more in rating, or a water-cooled system containing Group 1 refrigerant with a rating of 3 hp or more. A stop or shutoff valve will not be placed between a pressure-limiting device and the compressor it serves.

Sections 1515-1517 of the UMC should be consulted for various pressure-relief valve and pressure-relief device requirements.

Refrigerating systems containing a Group 2 refrigerant or carbon dioxide and located inside a building are required to have a means for manual discharge of the refrigerant into the atmosphere. These systems will also be equipped with manual means of releasing the refrigerant from the high-pressure side of the system to the low-pressure side.

6 Installation

Testing, Adjusting, and Balancing (TAB)

The following is a preliminary check list to aid in the preparation for TAB:

1. Obtain a piping flow diagram showing all equipment. Record flow rates and temperatures on the diagram.
2. Obtain all equipment data from the manufacturers, and from the design specifications.
3. Obtain and calibrate the instrumentation that applies best to each TAB task.
4. Decide where all measurements will be taken, and check to see if access is possible.
5. Make sure all valves and controls are in correct position.

The following is a basic TAB procedure to ensure proper flow through the chiller:

1. Check pump speed with condenser and chilled water design.
2. Slowly close the pump discharge balancing valve, recording discharge and suction heads, motor amps, and volts. Repeat this for various settings from valve fully closed to fully open. This information can be used to plot a pump performance curve. The pump head can then be corrected for any differences in velocity heads entering and leaving.
3. Adjust the system flow to approximately 110 percent of design GPM (according to pump curve).
4. Check and balance flow rates through large coils and chiller. Adjust balancing valves to within ± 10 percent of design GPM.
5. Check and balance flow rates to terminal units and adjust to within ± 10 percent of design GPM.
6. Repeat the balancing process until no change is found.
7. Measure and adjust water flow to cooling tower. Check performance of cooling tower by measuring water flow rate and temperatures, and air dry-bulb and wet-bulb temperatures in and out.
8. Carry out performance tests of chiller(s) and cooling tower(s) with the help of manufacturer's field engineers.

Acoustics

Because of their noise, it is highly advisable to locate chillers far away from any noise-sensitive area.

Two forms of vibration reduction—damping and isolation—can be used. Damping is accomplished by rigidly coupling the vibrating source to a large mass, often referred to as an inertia block. A great amount of the energy is absorbed and dissipated as friction. The remaining vibration results in lower-amplitude vibration. Isolation is accomplished by supporting the vibrating mass on resilient supports. Figure C-27 shows examples of vibration reduction applications.

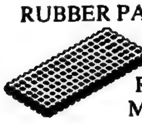
Machines can be supported on fibrous, rubber, or steel vibration isolators, and the entire mass can be supported on a floating floor that rests on resilient vibration isolators. Flexible joints in all pipes and ducts connected to a vibrating machine are mandatory.

Large machines are supported on special commercial "sandwiches" of lead, cork, and other resilient materials. Machines with a dominant vibrational frequency can have special springs designed to give maximum isolation and damping at that frequency. Massive machines and impacting devices use huge inertia blocks and even separate foundations to isolate their vibration.

Reciprocating and centrifugal chillers are among the mechanical equipment that require the most concern with respect to acoustical considerations. As is true with all mechanical equipment, the quality of noise data available from manufacturers varies tremendously. Some manufacturers perform exhaustive tests according to specific test standards and can provide very useful test data. With other manufacturers, caution should be exercised in looking at their data.

Reciprocating chillers are most commonly seen in applications requiring small cooling capacities, generally below 300 tons. The reciprocating motion of this chiller generates a great amount of noise and vibration. This makes the location of the equipment critical if noise intrusion into adjacent spaces is going to be a problem. These chillers are best installed only in slab-on-grade or basement locations where the vibration isolation can be accomplished easily. Even in basement locations, spring isolators will probably be required along with resilient flexible connections for all piping, electrical, and plumbing connections to the chiller.

RUBBER PADS (Type 1)



RUBBER MOUNTS (Type 2)

Note 20. Rubber isolators are available in pad (Type 1) and molded (Type 2) configurations. Pads are used in single or multiple layers. Molded isolators come in a range of 30 to 70 durometer (a measure of stiffness). Material in excess of 70 durometer is usually ineffective as an isolator. Isolators are designed for up to 0.5-in. deflection, but are used where 0.3-in. or less deflection is required. Solid rubber and composite fabric and rubber pads are also available. They provide high load capacities with small deflection and are used as noise barriers under columns and for pipe supports. These pad types work well only when they are properly loaded and the weight load is evenly distributed over the entire pad surface. Metal loading plates can be used for this purpose.

GLASS FIBER PADS (Type 1)

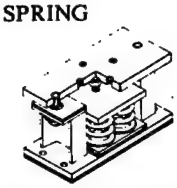
Note 21. Precompressed glass fiber isolation pads (Type 1) constitute inorganic inert material and are available in various sizes in thicknesses of 1 to 4 in., and in capacities of up to 500 psi. Their manufacturing process assures long life and a constant natural frequency of 7 to 15 Hz over the entire recommended load range. Pads are covered with an elastomeric coating to increase damping and to protect the glass fiber. Glass fiber pads are most often used for the isolation of concrete foundations and floating floor construction.

SPRING ISOLATOR (Type 3)

Note 22. Steel springs are the most popular and versatile isolators for HVAC applications because they are available for almost any deflection and have a virtually unlimited life. All spring isolators should have a rubber acoustical barrier to reduce transmission of high-frequency vibration and noise that can migrate down the steel spring coil. They should be corrosion-protected if installed outdoors or in a corrosive environment. The basic types include

1. **Note 23.** Open spring isolators (Type 3) consist of a top and bottom load plate with an adjustment bolt for leveling. Springs should be designed with a horizontal stiffness at least 100% of the vertical stiffness to assure stability, 50% travel beyond rated load and safe solid stresses.

RESTRAINED SPRING ISOLATOR (Type 4)



2. **Note 24.** Restrained spring isolators (Type 4) have hold-down bolts to limit vertical movement. They are used with (a) equipment with large variations in mass (boilers, refrigeration machines) to restrict movement and prevent strain on piping when water is removed, and (b) outdoor equipment, such as cooling towers, to prevent excessive movement because of wind load. Spring criteria should be the same as for open spring isolators, and restraints should have adequate clearance so that they are activated only when a temporary restraint is needed.

3. Housed spring isolators consist of two telescoping housings separated by a resilient material. Depending on design and installation, housed spring isolators can bind and short circuit. Their use should be avoided.

AIR SPRINGS

ROLLING LOBE

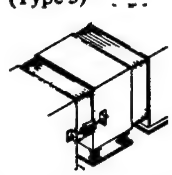
BELLOWS

Air springs can be designed for any frequency but are economical only in applications with natural frequencies of 1.33 Hz or less (6-in. or greater deflection). Their use is advantageous in that they do not transmit high-frequency noise and are often used to replace high deflection springs on problem jobs. Constant air supply is required, and there should be an air dryer in the air supply.

RUBBER HANGER (Type 2)

SPRING HANGER (Type 3)

THRUST RESTRAINT (Type 5)



Note 25. Isolation hangers (Types 2 and 3) are used for suspended pipe and equipment and have rubber, springs, or a combination of spring and rubber elements. Criteria should be the same as for open spring isolators. To avoid short circuiting, hangers should be designed for 20 to 35° angular hanger rod misalignment. Swivel or traveler arrangements may be necessary for connections to piping systems subject to large thermal movements.

Note 26. Thrust restraints (Type 5) are similar to spring hangers or isolators and are installed in pairs to resist the thrust caused by air pressure.

Figure C-27. Vibration Reduction Applications.

Reprinted with permission from the 1995 *ASHRAE Applications Handbook.*

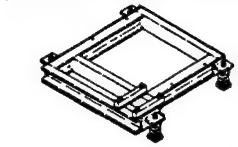
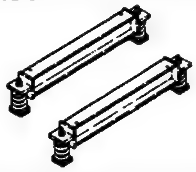
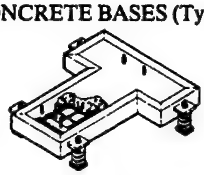
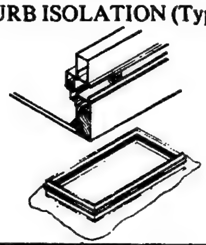
STRUCTURAL BASES (Type B) 	**Note 28.** Structural bases (Type B) are used where equipment cannot be supported at individual locations and/or where some means is necessary to maintain alignment of component parts in equipment. These bases can be used with spring or rubber isolators (Types 2 and 3) and should have enough rigidity to resist all starting and operating forces without supplemental hold-down devices. Bases are made in rectangular configurations using structural members with a depth equal to one-tenth the longest span between isolators, with a minimum depth of 4 in. Maximum depth is limited to 12 in., except where structural or alignment considerations dictate otherwise.
STRUCTURAL RAILS (Type B) 	**Note 29.** Structural rails (Type B) are used to support equipment that does not require a unitary base or where the isolators are outside the equipment and the rails act as a cradle. Structural rails can be used with spring or rubber isolators and should be rigid enough to support the equipment without flexing. Usual industry practice is to use structural members with a depth one-tenth of the longest span between isolators with a minimum depth of 4 in. Maximum depth is limited to 12 in., except where structural considerations dictate otherwise.
CONCRETE BASES (Type C) 	**Note 30.** Concrete bases (Type C) consist of a steel pouring form usually with welded-in reinforcing bars, provision for equipment hold-down, and isolator brackets. Like structural bases, concrete bases should be rectangular or T-shaped and, for rigidity, have a depth equal to one-tenth the longest span between isolators, with a minimum of 6 in. Base depth need not exceed 12 in. unless it is specifically required for mass, rigidity, or component alignment.
CURB ISOLATION (Type D) 	**Note 31.** Curb isolation systems (Type D) are specifically designed for curb-supported rooftop equipment and have spring isolation with a watertight and airtight curb assembly. The roof curbs are narrow to accommodate the small diameter of the springs within the rails, with static deflection in the 1- to 3-in. range to meet the design criteria described for Type 3.

Figure C-27. Vibration Reduction Applications (continued)

Because turbulence in the chilled water pipes can be significant, it is necessary in a majority of cases also to isolate the pipes from the supporting structure with spring hangers. If the chiller is located anywhere close to a sensitive area, care must be exercised in assuring that adequate isolation is provided.

For applications where over 300 tons of cooling is necessary, centrifugal chillers are generally chosen. These are either direct or gear driven machines. All centrifugal chillers have a smooth rotary motion to their operation, which generates far less vibration than the reciprocating chillers. Centrifugal chillers also generate less low-frequency noise than reciprocating chillers. The higher operating speed of centrifugal chillers puts the majority of their generated noise in the mid-frequency range.

7 Operation and Maintenance

Controls for Chillers

Chiller Plants

Chiller plants are generally specified as variable flow or constant flow (Figures C-28 and C-29). The type of control of the remote load is the determining factor. Remote loads with two-way control valves require variable flow, whereas remote loads with three-way valves permit constant flow.

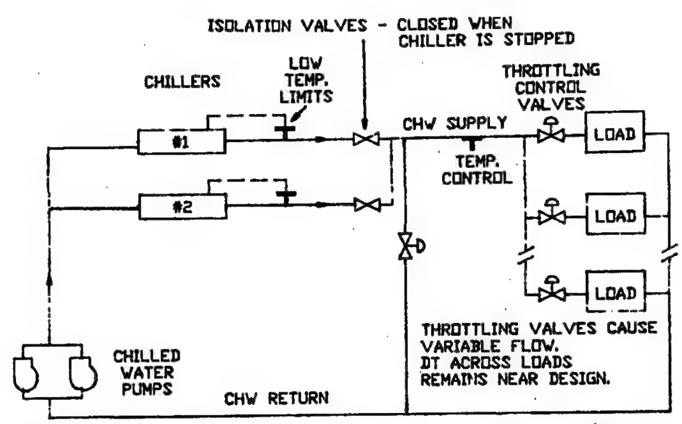


Figure C-28. Variable Flow Chilled Water System (Parallel Flow).
Reprinted with permission from the 1995 *ASHRAE HVAC Applications Handbook*.

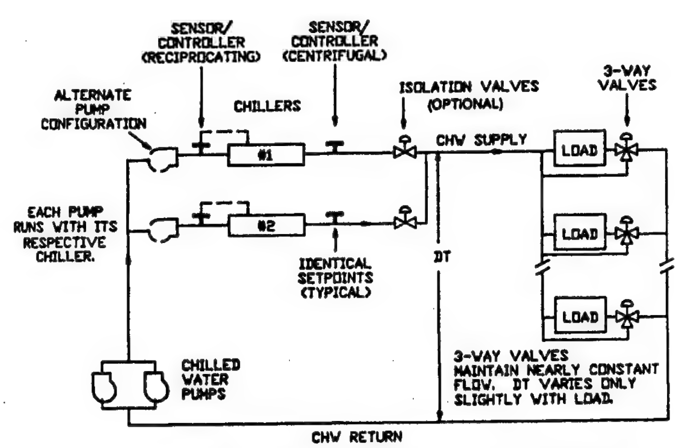


Figure C-29. Constant Flow Chilled Water System (Parallel Flow).
Reprinted with permission from the 1995 *ASHRAE HVAC Applications Handbook*.

Liquid Chillers

Some common controls for liquid chillers are:

- *Chilled Liquid Temperature Sensor:* Sends signal to the control circuit, which modulates compressor capacity.
- *Water Temperature Controller:* Cycles the compressor(s) when the cooling load drops below a minimum.
- *Anti-Recycle Timer:* Limits starting frequency.
- *Current Limiter:* Limits compressor capacity during periods of high power consumption.
- *High Pressure Cut-Out:* Pressure actuated switch used to protect the compressor from pressures caused by lack of water or air, or high condenser temperatures.
- *Low Pressure Cut-Out:* Pressure or temperature actuated device used in the evaporator to protect it from freezing chilled water. Direct-expansion systems cannot use this device.
- *High Oil Temperature:* Protects the compressor if loss of oil cut-out cooling occurs or if a bearing failure causes excessive heat generation.
- *Oil Failure Switch:* Shuts down the compressor if oil pressure drops below a minimum value, or if sufficient oil pressure is not developed shortly after compressor start-up.
- *High Motor Temperature Sensor:* Shuts down machine if loss of motor cooling, or overload because of a failure of operating controls occurs.
- *Low Oil Sump Temperature Switch:* Protects against an oil heater failure, or prevents starting after a prolonged shutdown before the oil heaters have had time to drive off refrigerant dissolved in the oil.
- *Chilled Liquid Flow Interlock Switch:* Protects external piping against a cooler freeze-up in the event of a liquid flow stoppage.

These devices may or may not be furnished with the liquid chilling package. The chiller type will determine which ones need to be used.

Centrifugal and screw chillers have a wide range of capacity through continuous modulation. Reciprocating chiller capacity reductions are limited to specific values. Temperature control is therefore set up differently in these systems.

Each central chiller plant is different, with different characteristics of sizes, manufacturers, chiller types, cooling towers, drives, piping configurations, and loads, so controls are custom designed for each plant.

Absorption

Absorption units usually come with a control panel containing electric motor starters, fuses, and other controls that can be grouped in a panel. A control voltage transformer and disconnect switch may also be included. These are all factory wired and installed. Additional controls commonly available are:

1. Condensing water-flow switch
2. Controls for series or parallel unit operation
3. Solution flow control
4. Automatic de-crystallization control
5. Steam demand limiter.

Some typical protective controls for absorption cooling are:

- *Low Temperature Cut-Out:* Thermostat that stops the unit when the evaporator temperature falls too low.
- *Cooling-Water Switch:* Stops the unit when the cooling water supply fails.
- *Chilled-Water Switch:* Stops the unit when the chilled water flow drops below design limits.
- *Concentration Limiters:* Sensors that indicate limiting conditions of absorbent concentration. They reset when the limiting conditions no longer exist.
- *Overflow Piping ("J" Loop):* Allows hot solution from the generator to overflow to the absorber in case another control fails.
- *Hot Water Cut-Out:* For machines with heating capabilities, temperature cutout limits the temperature and restarts the machine when an acceptable level is reached.
- *Protective Devices for Direct-Fired Machines:*
  - Low level switch (for solution)
  - High pressure switch
  - High temperature control
  - Flame ignition and monitoring control.

Computer Applications for Chillers

The use of computers as a management tool allows a building to be viewed as a complete system by monitoring loads and trends, and by rapidly providing operational profiles. The energy usage and performance of chiller components within a system can be monitored by a computer. The organized relationship of the coils

with the central chiller plant, and with one another, is the major influence in an energy conservation program.

The following is a basic evaluation procedure for a central processor managing a chiller:

1. Total all loads.
2. Total load is matched with a list of available sources to match current source capacity most efficiently with total load.
3. Downtime and demand considerations are taken into account when a list of available sources is being generated.
4. If downtime or demand considerations call for stopping a given chilled water source, the source is removed from the list until it is available again.
5. Start no source that has been recently stopped.
6. Stop no source that has been recently started.
7. Select sources by considering expected load changes based on time-of-day experiences and outside conditions.
8. Communicate start/stop requests to stand alone microprocessors.

In addition to operating the equipment, the following information could be obtained from the central processor:

- Building load vs. time of day
- Total load vs. time of day
- Building demand vs. time of day
- Total demand vs. time of day
- Individual source total operating time.

Maintenance

Reciprocating, Centrifugal, and Screw Chillers

As with the absorption chiller, the manufacturer's specific recommendations should be followed. Below is a list of general maintenance specifications that apply equally to reciprocating, centrifugal, and screw chillers. In general, equipment should be neither overmaintained nor neglected.

Continual Monitoring:

1. Condenser water treatment is determined specifically for the condenser water used.
2. Operating conditions - daily log sheets are recommended.

Periodic Checks:

1. Leak check
2. System dryness
3. Oil level
4. Oil filter pressure drop
5. Refrigerant quantity or level
6. System pressures and temperatures
7. Water flows
8. Expansion valves operation.

Regularly Scheduled Maintenance:

1. Condenser and oil cooler cleaning
2. Calibrating pressure, temperature, and flow controls
3. Tightening wires and power connections
4. Inspection of starter contacts and action
5. Dielectric checking of hermetic and open motors
6. Oil filters and drier change
7. Analysis of oil and refrigerant
8. Seal inspection
9. Partial or complete valve or bearing inspection
10. Manufacturer's recommendations.

Screw Chillers

Some maintenance for screw chillers differs substantially from reciprocating or centrifugal units, so special attention is given to the maintenance of the screw-type chiller.

Due to the large oil flows that are a part of the screw compressor system, the oil filter pressure drop should be monitored carefully and the elements changed periodically. This maintenance is very important in the first few months after

start-up of any factory-built package and is absolutely essential on field-erected systems.

The oil and refrigeration systems merge at the compressor. Therefore, much of the loose dirt and fine contaminants in the system eventually find their way to the oil sump, where they are removed by the oil filter. The filter-drier cartridges should also be monitored for pressure drop and moisture during initial start-up, and regularly in the future life of the unit. If a system reaches a certain level of dryness where the amount of moisture in the system is not detrimental to proper functioning of the machine, then the dryness level is acceptable. Oil does not have to be changed unless it becomes contaminated by water, acid, or metallic particles. It is good practice to check the oil for acidity periodically, using commercially available acid test kits.

Absorption

The length of a machine's useful life and the extent to which it performs trouble-free is related directly to the care taken in maintaining the unit's cleanliness and tightness. Internal cleanliness and leak tightness that last the life of the unit are the most important considerations for operation and maintenance.

Before absorption machines are sent out from the manufacturer, rigid standards of manufacture are ensured by testing the product with sophisticated equipment. Electronic halide leak detectors and helium mass spectrometers are used to ensure that the equipment has no leaks before shipment from the factory. Vacuum integrity and internal cleanliness are also checked.

In addition to the many initial checks and assurances of the manufacturer, the owner must do many things to assure a long life for the equipment. As is true in any case, the manufacturer's instructions should be followed for any specific piece of equipment.

Units built most recently will use corrosion inhibitors to help protect the internal parts from corrosive attack. Even though a corrosion inhibitor is used in the system, proper maintenance with regard to purging and leak tightness still must be used.

When a machine is opened to the atmosphere for repair and maintenance, nitrogen should be used to break the vacuum. Nitrogen is used because it is an inert

gas and will prevent corrosive attack on internal parts while they are exposed to the atmosphere.

Since purging is so important, the manufacturer's instructions on purge system maintenance should be followed, and the effectiveness of the purge system should be verified periodically. Leak tightness should also be evaluated periodically. All manufacturers describe procedures for measuring the leak rate, bubble count, or noncondensable accumulation rate of their machines. If the measured leak rate is excessive, it is important to find and repair the leak as soon as possible.

Internal cooling-water surfaces of the absorber usually require periodic cleaning by mechanical, chemical, or both means regardless of the effectiveness of the water treatment practices followed. The more effective the water treatment, the longer the allowable period between the cleaning.

Crystallization is the precipitation of salt crystals from absorbent. It is a slush-like mixture that can plug up pipelines and other fluid passages in the machine and cause it to stop operating. Crystallization used to occur in earlier models of absorption machines and may still occur in more recent models if they are not properly maintained. However, crystallization is not as common in the newer models.

Some models have a bypass pipe that will cause the solution to bypass the crystallization that usually occurs in the heat exchanger. The heat exchanger will naturally de-crystallize itself through the bypass process. Other manufacturers provide concentration control to avoid crystallization. The control of concentration is achieved by placing a device between the concentrator and the heat exchanger to sense and control the level of concentrate to the heat exchanger.

If a crystallization condition develops, the liquid level rises within the concentrated solution line as resistance to flow within the heat exchanger increases. The increase in level is sensed by the sensing device that opens the valve. The opening of the valve allows refrigerant to flow into the concentrated solution line, and reduces the solution concentrate. If crystallization does occur, it may be necessary to add water and heat to the part of the machine crystallized (usually the heat exchanger).

Crystallization does not harm the equipment, but it is a symptom of trouble and its cause should be found and corrected. The most common causes are controls

that are improperly set or malfunctioning, sudden drops in cooling-water temperature, atmospheric air leaks into the machine, and electrical power failure.

Fouling of external or machine-side tube surfaces of an absorption machine is not a problem because there is no continuous source of scale-forming substances such as algae. Nevertheless, as with all heat-transfer devices involving the heating of water inside tubes, it is necessary to use good water treatment practices on the internal or water side of the absorber and condenser tubes.

Water Treatment

Closed Systems

Closed cooling systems confine cooling water within the system pipes and heat exchanger. Although generally defined as systems that require less than 5 percent makeup per year, it is not uncommon to have a closed system loose up to 25 percent per month. Therefore, it is desirable to treat the water in these systems.

Corrosion and corrosion product deposit are the most serious problems encountered. Corrosion is a process in which metal returns to its original state. The actual disintegration of the metal will only happen at the anode. Cathode and anode areas can shift, which allows uniform corrosion. Water characteristics that influence the rate of corrosion are: temperature, water velocity, amount of suspended solids, degree of acidity, presence of microbial growths, and the presence of O_2.

When corrosion happens, corrosion products can build up in the system and form deposits throughout the heat exchanger and piping. Glycol and alcohol solutions are commonly used in closed systems to prevent corrosion. Chromate inhibitors are the most effective, but cannot be used in locations where drainage water can cause pollution. Softeners should be added to makeup water to prevent scale.

Open Systems

Open circulating systems include cooling towers, evaporative condensers, or spray ponds. Water is continuously reused in these systems, but is also exposed to air. Makeup water must continuously be added to replace evaporated water and water lost through leaks.

Cooling tower problems are commonly classified into three categories: deposit formation, corrosion, and biological deposition.

Deposit formation. Deposit formation includes fouling and scale. Fouling is the accumulation of water-suspended materials on heat-exchanger surfaces. Scale is a coating of mainly inorganic materials, and results from supersaturation of water-soluble minerals. Deposits of both types reduce efficiency by reducing the rate of heat transfer within the heat exchanger. Fouling also aides in equipment deterioration.

Reduction of scale forming particles can be accomplished by lime-soda softening, ion exchange, or reverse osmosis. Solubilizing chemicals are also used to keep scale-forming materials in solution. The most commonly used are polymeric organics and organic phosphorus compounds. Crystal modifiers can also be used. These modifiers allow scale to form, but then deform the crystal structure, changing the scale into sludge. Sludge usually does not build up on heat-exchange surfaces. Polymaleic acids and sulfonated polystyrene are both effective modifiers.

Filters are widely used on cooling waters to remove foulants. Another technique for controlling fouling is injecting small rubber balls into and through heat exchanger tubes during operation, which wipes the tubes clean as they pass through. Chemical treatment is also used in controlling fouling. Fouling conditions and the particular foulant involved must be properly matched, as no one chemical is known to work on all foulant control problems.

Corrosion. Noncorrosive metals can be used in the design of a system to help minimize corrosion. Paint, epoxy, or metal plating can also be applied to help reduce corrosion. Another method is the application of cathodic-protection, which uses induced electrical currents.

Chemical inhibitors are another highly used technique, which allow a protective film to be formed over the metal. Inhibitors are fed into the cooling water and transported to the metal surfaces. Cathodic inhibitors interfere with cathodic reactions. These inhibitors reduce the corrosion rate in direct proportion to the reduction of the unprotected cathodic area. Anodic inhibitors are often considered dangerous. If used incorrectly, severe corrosion can occur in small, unprotected areas. This can lead to the metal perforating in a very short time. General corrosion inhibitors can also be used, which will protect both anodic and cathodic surfaces.

Biological deposition. Biological deposition includes macrobiological fouling of discharge and intake canals, and microbiological fouling of heat exchangers. When algae, bacteria, fungi, and other organisms enter a system, they search for the best environment suited to their growth. Treatment against these falls into three main categories: chemical, mechanical, and thermal.

Chemical treatment is very effective in dealing with biological deposition. Regulated limitations have been set on this type of treatment due to the increasing concern for environmental effects. Temperature, pH, system design, and limitations on the discharge of toxic substances should be considered when selecting a chemical treatment.

Mechanical methods include rakes, trash bars, and strainers. Physical cleanup is usually the most effective mechanical approach. This involves the use of brushes, scrapers, and sponge balls in the cleanup of accumulated inorganic foulants.

Thermal methods involve temperature elevation, which is based on the fact that some organisms cannot survive when exposed to extreme temperatures. This approach is often combined with a thermal-backwash procedure, in which reversible gates are used to alternate coolant-flow direction, which gives a flushing action.

8 Acceptance Testing

When testing a chiller, it is important for the instruments used in testing the equipment to be calibrated and functioning properly. It is also important that the testing personnel understand the functions of the instruments, and how to operate and collect data from them.

A report worksheet that lists all the data that need to be collected to analyze the operation of the chiller should be used by the testing personnel. The following worksheet contains four sections. Each section requires design data to be recorded and actual operating data to be collected. This allows a comparison to be made between the two sets of data. If the actual operating data differ significantly from the design data, the actual data should be measured again to make sure the test instruments are functioning properly. If the test instruments are found to be working properly and the actual data is still significantly different, then there is a need to troubleshoot the chiller to determine the reason for the discrepancy.

CHILLER SYSTEM ACCEPTANCE TESTING CHECKLIST

PROJECT:____

LOCATION:_______________________________________

NAME:____

A. Evaporator		Correct yes	Correct no	Date Checked
1. Installed gauges and controls				
2. Entering/leaving water temperature				
3. Water temperature, ΔT				
	Design	TAB	Actual	
4. Correct flow (gpm)				

B. Condenser		Correct yes	Correct no	Date Checked
1. Installed gauges and controls				
2. Entering/leaving water temperature				
3. Water temperature, ΔT				
	Design	TAB	Actual	
4. Correct flow (gpm)				

C. Compressor	Correct yes	Correct no	Date Checked
1. Installed gauges and controls			
2. Make/model			
3. Serial number			
4. Voltage: T_1-T_2, T_2-T_3 T_3-T_1			
5. Amps: T_1, T_2, T_3			

D. Pumps-Motors	Correct yes	Correct no	Date Checked
1. Make, model numbers, etc.			
2. Clean and free of foreign objects			
3. Rotation			
4. Lubrication			
5. Alignment/securely fastened			
6. Guards in place			
7. Pressure gauges installed			
8. Power available			
9. Disconnects installed and labeled			
10. Interlocks functional			

E. Pumps-Piping	Correct yes	no	Date Checked
1. Flexible connectors			
2. Connections			
3. Pressure and temperature at pump inlet			
4. Air bled from casing where required			
5. Free of leaks			
6. Strainer clean			
7. Air vented			
8. Piping system pressure tested			
9. Pipes labeled			
10. Valves tagged			
11. Chemical treatment system installed			
12. Water treatment report submitted			
13. "TAB" complete and approved			
14. Correct flow (gpm)			

F. Cooling Tower/Evaporative Condenser	Correct yes	no	Date Checked
1. Correct flow and connections			
2. Valves open or set			
3. Leakage			
4. Provisions made for "TAB" measurements			
5. Sump water level			
6. Spray nozzles			
7. Fan/pump rotation			
8. Motor/fan lubrication			
9. Drives and alignment			
10. Guards in place			
11. Starters and disconnect switches			
12. Electrical connections			
13. Nameplate data			

NOTES

- Compressor: Nameplate information may be checked to see that the proper chiller was installed.

- The voltage, amps, and kW input may be measured and evaluated by use of a voltammeter, which is an electrical, clamp-on type measuring device. With this device, transformer currents may be read without interrupting electrical services. When using this device, safety precautions must be taken. These precautions and testing procedures are found in many texts (SMACNA 1983; NEBB 1984).

- To measure voltage, set the meter to the most reasonable range, connect the test lead probes firmly against the terminals or other surfaces of the line being tested, and read the meter. Be sure to read the correct scale if the meter has more than one scale. When reading single-phase voltage, the leads should be applied to the two load terminals. When reading three-phase current, it is necessary to apply the probes to the terminal poles No. 1 and No. 2; then to poles No. 2 and No. 3; and finally to poles No. 1 and No. 3. The three readings that are obtained will probably be slightly different but close to each other. For practical purposes, the readings may be averaged. From the design data, determine if the voltage is high or low.

Glossary

ACOUSTIC VELOCITY: Velocity of sound; the practical velocity of a gas through openings or in piping is limited by this velocity.

BTU (BRITISH THERMAL UNIT): Energy; the amount of heat required to raise one pound of water by one degree F.

BTU/H: Power; used to express the total heat loss or gain of a building.

COEFFICIENT OF PERFORMANCE (COP): A factor that measures refrigeration requirements with power input.

COMPRESSOR: The pump in a mechanical refrigeration system that compresses the refrigerant vapor into a smaller volume, thereby raising the pressure of the refrigerant and consequently its boiling temperature. The compressor is the separation between the high and low side.

DAMPING: Diminishing by some means the activity caused by the introduction of energy to a system.

DOWNTIME: The increment of time a system or component of a system is not functioning or being utilized.

HEAD: Dynamic or total; in flowing fluids the sum of the static and velocity heads at the point of measurement.

HEAD, STATIC: The static pressure of fluid expressed in terms of the height of a column of fluid.

HEAD, VELOCITY: In a flowing fluid, the height of the fluid or of some manometric fluid equivalent to its velocity pressure.

HEAT EXCHANGER: A device specifically designed to transfer heat between two physically separated fluids.

MODULATION: Of a control, tending to adjust by increments and decrements.

NATURAL CONVECTION: Circulation of gas or liquid (usually air or water) due to differences in density resulting from temperature changes.

POSITIVE DISPLACEMENT COMPRESSOR: Increases the pressure of the refrigerant vapor by reducing the volume of the compression chamber by a fixed amount. Includes reciprocating and screw compressors.

PURGE: To free of sediment or relieve trapped air by bleeding.

REFRIGERANT, ABSORPTION CYCLE: Distilled water.

REFRIGERANT, REFRIGERATION CYCLE: A volatile substance that absorbs heat from a place where it is not wanted, and disposes of it in another place.

RESILIENCE: Ability to recover from or adjust to deformation usually caused by compression.

TESTING, ADJUSTING, AND BALANCING (TAB): Testing, adjusting, and balancing a system.

THERMAL EXPANSION VALVE: The metering device or flow control that regulates the amount of liquid refrigerant which is allowed to enter the evaporator.

TON, REFRIGERANT: One ton of refrigerant is the amount of cooling required to change one ton of water into one ton of ice over a 24 hour period.

VAPOR: A gas, particularly one near equilibrium with the liquid phase of the substance, and which does not follow the gas laws.

VARIABLE DISPLACEMENT COMPRESSOR: Uses force to raise the pressure of the refrigerant. Includes centrifugal compressors.

VIBRATION: An oscillatory motion generated by the introduction of energy to the piece affected.

Bibliography

American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), *Applications Handbook*, ASHRAE, Atlanta, GA, 1995.

ASHRAE, *Handbook of Fundamentals*, ASHRAE, 1997.

ASHRAE, *HVAC Systems and Equipment Handbook*, ASHRAE, 1996.

ASHRAE, *Products Specification File*, ASHRAE, 1982.

ASHRAE, *Refrigeration Handbook*, ASHRAE, 1998.

Bradshaw, Vaughn, *Building Control Systems, Second Edition*, John Wiley & Sons, New York, New York, 1993.

Clark, Earl M.; Bates, Robert L.; Anderson, George G.; Wells, Ward D., "Retrofitting existing chillers with alternative refrigerants," *ASHRAE Journal*, April 1991

Moyer, James A., Fittz, Raymond U., *Air Conditioning*, McGraw-Hill, New York, 1975.

Pita, Edward G., *Air Conditioning Principles and Systems*, John Wiley & Sons, New York, 1989.

Stein, Benjamin; Reynolds, John S.; McGuinness, William J, *Mechanical and Electrical Equipment for Buildings, 7th Edition*, John Wiley & Sons, New York, 1986.

Strauss, Sheldon and Puckorius, Paul, "Cooling-Water Treatment," *Power Magazine*, June 1944.

Trane Company, *Absorption Refrigeration*, LA Crosse, Wisconsin, 1972.

Uniform Mechanical Code, Chapters 13, 14, 15, Whittier, California, 1988.

Appendix D: Exhaust Systems

Principles, Applications, and Acceptance Testing

by
Dahtzen Chu
Energy Branch, Facilities Division
U.S. Army Construction Engineering Research Laboratory

and

Charles L. Burton
Leland V. Speirs
Department of Architectural Engineering and Construction Science
Kansas State University

Contents

1 Introduction

Outdoor air that flows through a building, either unintentionally as infiltration or intentionally as ventilation, is important for two reasons:

1. Outdoor air is usually used to dilute indoor air contaminants.
2. Energy associated with heating or cooling outdoor air is a significant space-conditioning load.

Buildings use this air movement in three different modes for air exchange:

1. Forced-air ventilation
2. Natural ventilation
3. Infiltration.

These modes affect air quality, energy, and thermal comfort very differently. They also have different capabilities in maintaining a desired air exchange rate. All three modes should be included in the air exchange rate in a building at any given time.

Infiltration is the uncontrolled flow of air through unintentional openings in a building's envelope or shell. It can be driven by pressure differences across the shell (i.e., by appliance-induced pressures, temperature differences, and wind). It is an important factor in mechanically ventilated buildings.

Natural ventilation is caused by pressures from indoor-outdoor temperature differences and pressures from wind. Air flow through open windows and doors can be used to provide adequate ventilation for contaminant dilution and temperature control in some cases. In other cases, unintentional openings in the building envelope can interfere with desired natural ventilation air distribution patterns and lead to larger-than-design air flow rates.

Forced-air ventilation depends on the air flow rates through the system fans, the air flow resistance associated with the air distribution system, the air flow resistance between the zones of the building, and the air-tightness of the building

envelope. Forced-air ventilation has the greatest control of air exchange rate and air distribution within a building. It is generally mandatory in larger buildings, where a minimum amount of outdoor air is required for occupant health and comfort and where a mechanical exhaust system is necessary.

2 Infiltration

Infiltration is the uncontrolled flow of air through openings in a building's envelope driven by pressure differences across the shell.

Air flow through the building shell occurs because pressure differences act on openings in the shell. Understanding infiltration requires understanding the pressures that cause the flow and the flow characteristics of the openings in the building shell.

Building pressure is determined by how much air is being introduced compared to how much is being exhausted. If more air is introduced than is exhausted, the difference should pressurize the building and leak out through cracks. Therefore, the size of the cracks, or "porosity" of the structure, is an important factor in building pressurization.

Factors Determining Building Pressure

Porosity and several other factors can affect building pressure. Some of these factors are controllable while others are not.

Building Porosity

Building porosity is composed of many variables:

- Leakage through doors
- Leakage through windows (movable and stationary)
- Leakage through elevator shafts
- Leakage through walls
- Building age—porosity will change as a building settles

- Leakage through fireplace dampers
- Leakage across the top ceiling of the heated space.

Stack Effect

Stack effect occurs when the temperature inside a building is not equal to the outdoor air temperature. Flow within the building results from the pressure differences that occur due to the differences in the air density.

The stack effect is most noticeable in multistory buildings when outdoor air temperatures are considerably less than indoor air temperatures. This results in pressure differences of some magnitude between upper and lower floors. Upper floors are of a positive pressure relative to the atmosphere while the lower floors are negative. The result is an upward air flow, generally through the elevator shafts and stair wells.

The reverse will occur during the summer when the indoor temperature is less than the outdoor temperature, but the effect will be reduced if the temperature and corresponding pressure differential between indoors and outdoors is low. During the cooling season, the temperature difference generally is not greater than 30 °F compared to a possible 80 °F temperature difference during the winter. Therefore, the infiltration of air in summer is at the upper floors and the exfiltration of air at lower floors. Resulting air flow is down through the building and is minimal.

Control in the past has been provided through isolation of elevator shafts (the most common carrier of air), building entrances, and the pressurization of first floor lobbies.

Wind Velocity and Direction

Wind velocity and direction tend to be uncontrollable factors. Air flow due to wind around or over a building will create areas in which static pressure will be different than the pressure of the undisturbed air flow (Figure D-1).

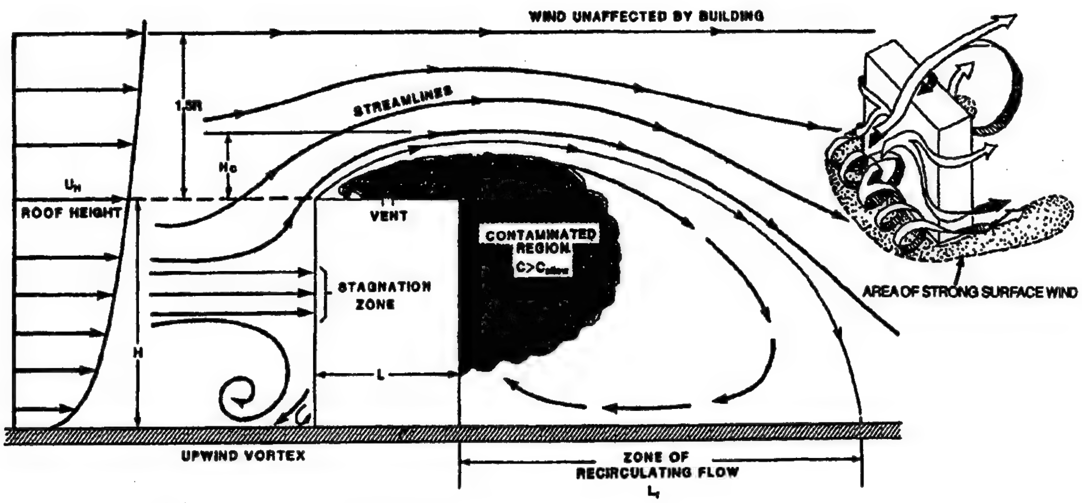


Figure D-1. Wind Velocity and Direction.
Reprinted with permission from the 1997 *ASHRAE Handbook.*

Pressures on the windward side of the building will be positive; negative on the leeward side. On the remaining sides, static pressures will be positive or negative to lesser degrees depending upon the direction of air flow.

The terrain surrounding the building can also create wind flow changes affecting building pressures. Surface roughness of the surrounding terrain (the size and location of surrounding buildings) influences the relationship of wind velocity to building height, which will affect the pressure patterns around the building's exterior, including the roof.

Moisture and Infiltration

Buildings, like our bodies, exchange moisture and air with the environment, as well as exchanging heat. Although most of this moisture exchange occurs during the exchange of fresh air, some exchange occurs through a building's skin. This can cause problems in either hot, humid climates or very cold ones.

In hot, humid conditions, cool inside surfaces are often encountered—for example, the ceiling directly below a roof pond used for passive cooling. As hot and humid air contacts such a surface, condensation can occur. The moisture vapor in the air condenses to form visible droplets of water on the ceiling. The result can be mildly annoying water drips on the head, or serious water stains, eventually with mold growing on surfaces.

In cold climates, cold interior surfaces also occur, especially at windows. Although the air indoors may not be particularly humid (40 to 50 percent RH is

common), it contains enough moisture to permit condensation on cold surfaces. Again, mild annoyance or more serious damage can result. A much less visible moisture threat occurs within walls, ceilings, or floors. Almost all common building materials, including gypsum board, concrete, clay masonry, and wood, are easily permeated by moisture. Most surface finishes are also permeable. In cold climates, the air outside contains relatively little moisture, even though the RH may be high. By contrast, inside air contains much more moisture per unit of volume, despite its probably lower RH. The result is a flow of vapor from high vapor pressure to low vapor pressure (typically warm to cold).

Such a flow occurs when the temperature within the wall (floor, etc.) drops low enough for this vapor to condense. Insulation can then become wet and thereby less effective, since water conducts heat far better than the air pockets it has filled. If wet insulation compacts, these air pockets are permanently lost. Worse yet, moisture damage can occur, such as dry rot in wood structural members. The usual remedy for such a potential problem is to install a vapor barrier within the building envelope. These barriers are commonly made of plastic film installed with as few holes as possible.

A substantial benefit of plastic films is that they reduce air flow through construction. Outdoor air is always infiltrating a building, gradually replacing the indoor air. This unintentional source of fresh air becomes a problem when temperatures outside are very different from those inside, especially when strong winds force outdoor air indoors fast enough to produce noticeably cold (or hot) drafts. Some fresh air is always desirable in buildings, but so is user control of how and where it is admitted. Therefore, the moisture-tight and infiltration-tight characteristics of plastic film vapor barriers are usually beneficial. When good vapor barriers are installed, the smaller air-change values that accompany "tight" construction may be assumed in calculations of heat flow due to infiltration and ventilation.

Methods of Calculating Infiltration

Air Change Method

The equation:

$$Q = \frac{(ACH) \times (room\ volume)}{60\ \mathrm{min}/h} = cu\ ft/\mathrm{min} \qquad \text{[Eq D-1]}$$

is used in calculating cubic feet per minute of infiltration air. In this equation, Q is the volume flow rate of air being calculated, and ACH is the number of air changes per hour expected, based on the type of construction (tight, medium, or loose) under the given conditions. Table D-1 is used in selecting values of ACH (ASHRAE 1979).

Example:

The infiltration of a room with dimensions 30 × 60 × 16 ft must be determined as part of a heat load calculation for winter time. The outside temperature is 0 °F with a 15 mi/h wind. The type of construction is medium.

Solution:

First, refer to Part B of Table D-1, and locate medium construction at 0 °F; the given value is 1.1 ACH. Inserting this value and the dimensions into the given equation provides the solution.

$$Q = \frac{(ACH) \times (roomvolume)}{60\ \text{min}/h} = \frac{(1.1) \times (30 \times 60 \times 16)}{60\ \text{min}/h} = 528\ cu\ ft/\text{min}$$

So, 528 cu ft/min of 0 °F air is entering this particular room.

Crack Method

The crack method assumes that data on wind velocities are known. It also assumes the doors and openable windows represent all the cracks by which outside air infiltrates a closed room under worst conditions. The following procedure is used to determine the infiltration.

The letter *k* represents the values for "window fit" and "door fit." These values are obtained from Part C of Table D-2 (ASHRAE 1979). *k* values are based upon tight, average, or loose fitting doors and windows.

Determine the outside average wind velocity in miles per hour and use Part A of Table D-2 to get the Velocity Head Factor (VHF). With the VHF, go to Part B of Table D-2 and use the VHF and *k*-curve to get the infiltration rate in cfm/ft.

Part A. Construction Types

Construction Type	Description
Tight	New buildings where there is close supervision of workmanship and special precautions are taken to prevent infiltration. Descriptions for tight windows and doors are given in Table 4.21.
Medium	Building is constructed using conventional construction procedures. Medium-fitting windows and doors are described in Table 4.21.
Loose	Buildings constructed with poor workmanship or older buildings where joints have separated. Loose windows and doors are described in Table 4.21.

Part B. Design Infiltration Rate (ACH) for Winter; Heating; Wind Speed = 15 mph

Type of Construction	Winter Outdoor Design Temperature (F)									
	50	40	30	20	10	0	−10	−20	−30	−40
Tight	0.4	0.5	0.6	0.6	0.7	0.8	0.8	0.9	0.9	1.0
Medium	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.2	1.3	1.4
Loose	0.8	0.9	1.0	1.2	1.3	1.4	1.5	1.6	1.8	1.9

Part C. Design Infiltration Rate (ACH) for Summer; Cooling; Wind Speed = 7.5 mph

Type of Construction	Summer Outdoor Design Temperature (F)					
	85	90	95	100	105	110
Tight	0.3	0.3	0.3	0.4	0.4	0.4
Medium	0.4	0.4	0.5	0.5	0.5	0.6
Loose	0.4	0.5	0.6	0.6	0.7	0.8

Part D. Infiltration per Square Foot of Floor Area

Ceiling Height (ft)	Air Changes per Hour																	
	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0
	cfm/ft²																	
7.5	0.04	0.05	0.06	0.08	0.09	0.10	0.11	0.13	0.14	0.15	0.16	0.18	0.19	0.20	0.21	0.23	0.24	0.25
8	0.04	0.05	0.07	0.08	0.09	0.11	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.21	0.23	0.24	0.26	0.27
8.5	0.04	0.06	0.07	0.09	0.10	0.11	0.13	0.14	0.16	0.17	0.18	0.20	0.21	0.23	0.24	0.26	0.27	0.28
9	0.05	0.06	0.08	0.09	0.11	0.12	0.14	0.15	0.17	0.18	0.20	0.21	0.23	0.24	0.26	0.27	0.29	0.30
	Btu/h ft² F																	
7.5	0.04	0.05	0.07	0.08	0.09	0.11	0.12	0.14	0.15	0.16	0.18	0.20	0.20	0.22	0.23	0.24	0.26	0.27
8	0.04	0.06	0.07	0.09	0.10	0.12	0.13	0.14	0.16	0.17	0.19	0.22	0.22	0.23	0.24	0.26	0.27	0.29
8.5	0.05	0.06	0.08	0.09	0.11	0.12	0.14	0.15	0.17	0.18	0.20	0.23	0.23	0.24	0.26	0.28	0.29	0.30
9	0.05	0.06	0.08	0.10	0.11	0.13	0.15	0.16	0.18	0.19	0.21	0.24	0.24	0.26	0.28	0.29	0.31	0.32

Table D-1. Estimated Overall Infiltration Rates for Small Buildings.

Source: *Mechanical and Electrical Equipment for Buildings*, 7th Ed., Stein, Reynolds, and McGuinness, 1986. Reprinted by permission of John Wiley & Sons, Inc.

Next, determine the linear feet of "crack" (LFC). The following example illustrates how to obtain the LFC. Using the equation:

$$Q = (\text{LFC}) \times (\text{infiltration rate}) \qquad \text{[Eq D-2]}$$

determine the infiltration in cfm.

Example

Consider the same room as used in the air-change method. Two opening windows are the only exterior openings for the enclosure (sizes are given in Figure D-2).

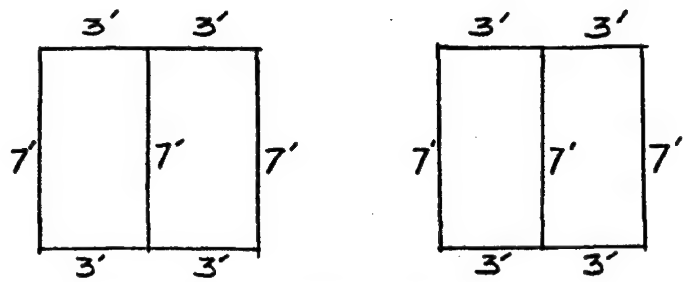


Figure D-2. Linear Feet of Crack of Windows.

From examining the windows and reading Part C of Table D-2, k = 2.0. The average wind speed for winter is found to be 15 mph.

Solution

Using Part A of Table D-2 with a wind velocity of 15 mph, the VHF is found to be 0.105.

With the obtained VHF, go to Part B of Table D-2 and obtain an approximate 0.49 (cfm/ft of crack) infiltration rate.

Next, determine the LFC of the two windows. The windows are identical in size and shape, so find the LFC for one window and multiply by 2.

LFC × 2 = (7' + 7' + 7' +3' + 3' + 3' + 3') = 66 ft of crack

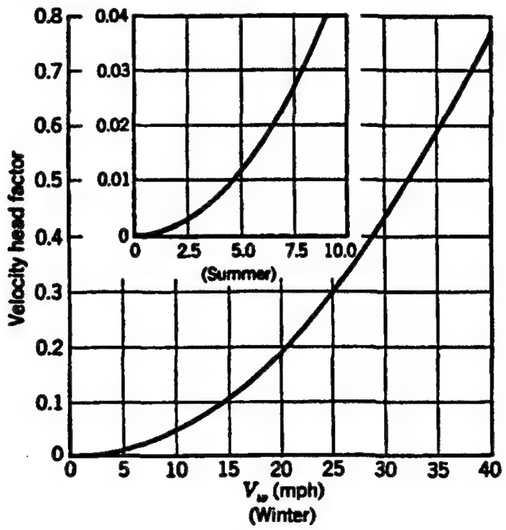
Next, substitute values into the given equation:

Q = (LFC) × (infiltration rate) = (66) × (0.49 cfm/ft) = 32.2 cu ft/min

Notice that for the same room, the air-change method estimated 528 cu ft/min was entering the room. In comparison to the crack method, there is a difference of 496 cu ft/min. It is important to select the method most appropriate for the existing circumstances.

Part A. Converting Wind Speed to Velocity Head Factor

NOTE: Typical design assumptions:
Winter wind V_w = 15 mph = velocity head factor of 0.105
Summer wind V_w = 7.5 mph = velocity head factor of 0.028



Part B. Infiltration Rates for Velocity Head Factors

NOTE: Enter this graph with velocity head factor (from Part A) to find infiltration rate in cfm/ft of crack (using values of *k* found in Part C or D).

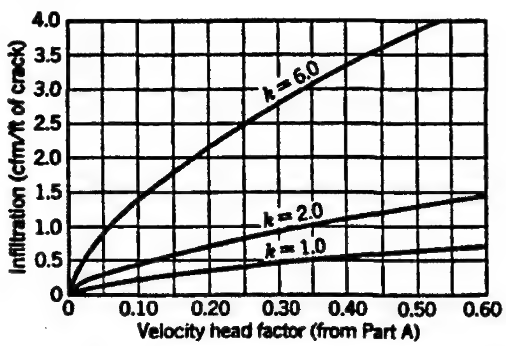


Table D-2. Approximate Infiltration Through Doors and Windows of Small Buildings.

Source: *Mechanical and Electrical Equipment for Buildings,* 7th Ed., Stein, Reynolds, and McGuinness, 1986. Reprinted by permission of John Wiley & Sons, Inc.

Part C. Classifications of Windows for Infiltration

Window Fit	Wood Double-Hung (Locked)	Other Types
Tight, k = 1.0	Weather-stripped; average gap (1/64-in. crack)	Wood casement and awning windows; weather-stripped.
		Metal casement windows; weather-stripped.
Average, k = 2.0	Nonweather-stripped; average gap (1/64-in. crack) or weather-stripped; large gap (3/32-in. crack)	All types of vertical and horizontal sliding windows; weather-stripped. If average gap (1/64-in. crack), this could be tight-fitting window.
		Metal casement windows; non-weather-stripped. If large gap (3/32-in. crack), this could be a loose-fitting window.
Loose, k = 6.0	Non-weather-stripped; large gap (3/32-in. crack)	Vertical and horizontal sliding windows; non-weather-stripped.

Part D. Classification of Residential-type Doors for Infiltration

Door Fit	Comments
Tight, k = 1.0	Very small perimeter gap and perfect fit weather-stripping—often characteristic of new doors
Average, k = 2.0	Small perimeter gap having stop trim fitting properly around door; weather-stripped
Loose, k = 6.0	Large perimeter gap having poor fitting stop trim; weather-stripped or Small perimeter gap; no weather-stripping

Table D-2. Approximate Infiltration Through Doors and Windows of Small Buildings (cont'd).

Curtain Wall Method

For this method, the amount of entering air is based on the wind blowing straight at an exposed wall of the room. Once again, construction classifications are used to determine values of k. These values of k are representative of tight, average, or loose fitting walls as designated in Table D-3.

Obtain the wind velocity and use Figure D-3 to determine velocity head in the form of $\Delta P_w/C_p$. ΔP_w is the change in pressure (inches of water), and C_p is the pressure coefficient for curtain wall buildings.

Leakage Coefficient	Description	Curtain Wall Construction
$k = 0.22$	Tight Fitting Wall	Constructed under close supervision of workmanship on wall joints. When joint seals appear inadequate they must be re-done.
$k = 0.66$	Average Fitting Wall	Conventional construction procedures are used.
$k = 1.30$	Loose Fitting Wall	Poor construction quality control or an older building having separate wall joints.

Table D-3. Curtain Wall Classification.

Reprinted with permission from *ASHRAE Cooling and Heating Load Calculation Manual,* 2nd ed.

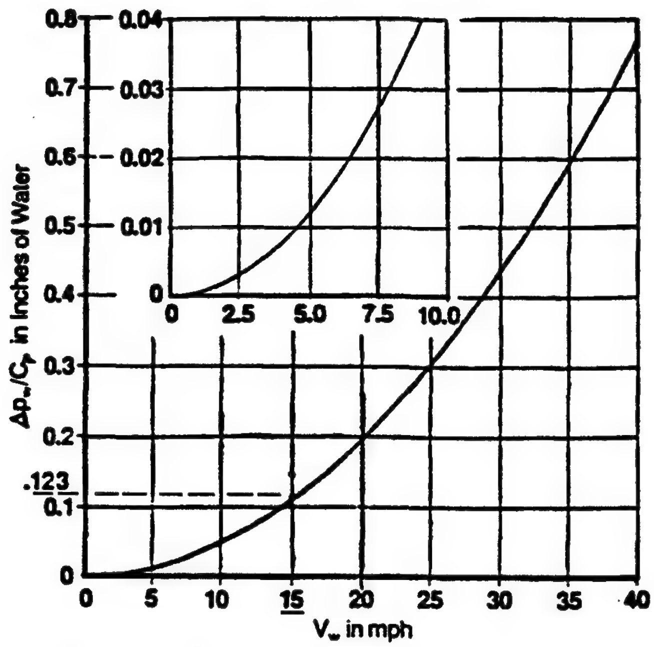


Figure D-3. Velocity Head vs. Wind Velocity.

Reprinted with permission from *ASHRAE Cooling and Heating Load Calculation Manual,* 2nd ed.

Then go to Table D-4 to get the C_p value. These values are determined by the direction of the wind (windward, leeward, and sides). By multiplying $\Delta P_w/C_p$ by C_p, the value of ΔP is obtained.

With ΔP, use Figure D-4 to obtain the air flow per square foot, $Q/A \; in \; cfm/sq \; ft$. The square foot area of the curtain wall under construction is then calculated and inserted into the equation $Q = A \times (Q/A) = cfm$.

The table is for a rectangular floor-shaped building and for wind normal to windward side.	
	C_p
Windward	0.95
Leeward	-0.15
Sides	-9.40

Table D-4. Wind Pressure Coefficients for Curtain Wall Buildings.

Reprinted with permission from *ASHRAE Cooling and Heating Load Calculation Manual*, 2nd ed.

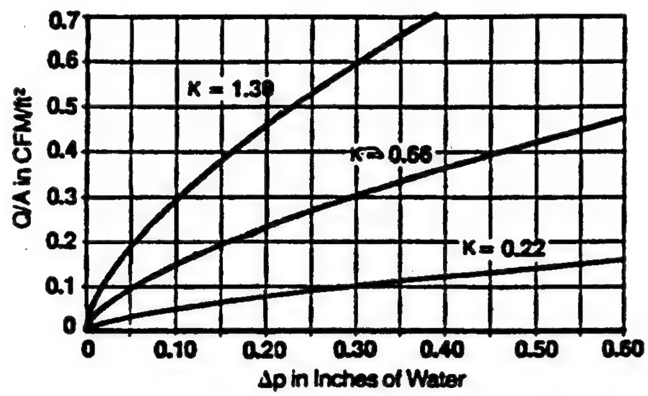


Figure D-4. Curtain Wall Infiltration for One Room or One Floor.

Reprinted with permission from *ASHRAE Cooling and Heating Load Calculation Manual*, 2nd ed.

Example

GIVEN: $k = 0.66$ (average)
wind velocity = 15 mph
$C_p = 0.95$ (windward)

FIND: Infiltration through the 60 x 16 ft wall of the room used in the two preceding examples.

Solution

$\Delta P_w/C_p = 0.123$ in. of water (Figure D-3)

$\Delta P = (\Delta P_w/C_p) \times C_p = (0.123) \times (0.95) = 0.1169$

Q/A = 0.175 cfm/sq ft (Figure D-4)

Area of Wall = 60 x 16 ft = $960\, sq\, ft$

$Q = A \times (Q/A) = 960\, sq\, ft \times (0.175$ cfm/sq ft) = 168 cfm

Stack Effect Method

When there is a difference in height between inlet openings situated low in the wall (or in floors) and outlets through roofs, and when outdoor air is cooler than indoor air, natural ventilation will occur through the stack effect of warm air rising and leaving through the higher openings.

The equation:

$$Q = C \times A \times \frac{h \times (t_i - t_o)}{t_i} \qquad \text{[Eq D-3]}$$

is used in calculating infiltration due to the stack effect. In this equation:

Q = air flow (cfm)

C = constant of proportionality = 313 (This assumes a value of 65 percent of the maximum theoretical flow, due to limited effectiveness of actual openings. With less favorable conditions, due to indirect paths from openings to the stack, etc., the effectiveness drops to 50 percent, and C = 240.)

A = area of cross-section through stack or outlets $(sq\ ft)$
Note: Inlet area must be at least equal to this amount.

t_i = (higher) temperature inside (°F), within the height h

t_o = (lower) temperature outside (°F)

h = height difference between inlets and outlets.

Example

Openings to the outside are indirect to the stack of a building; therefore, C=240. The cross-sectional area through the stack was measured to be 1.5 sq ft. Outlets are measured to be 15 ft above inlets. An outside temperature of 0 °F is measured. The inside temperature is 74 °F. Determine the amount of air entering the building due to the stack effect.

Solution

$$Q = C \times A \times \frac{h \times (t_i - t_o)}{t_i}$$

$$Q = 240 \times (1.5\,sq\,ft) \times \frac{15\,ft \times (74°F - 0°F)}{74°F}$$

$$= 1394\,cu\,ft/\text{min}$$

This is approximately three air changes per hour for the room used in the previous examples.

Natural Ventilation Guidelines

Several general guidelines should be followed when designing for natural ventilation:

- In hot, humid climates, maximize air velocities in the occupied zones for bodily cooling. In hot, arid climates, maximize air flow throughout the building for structural cooling, particularly at night when temperatures are low.
- Take advantage of topography, landscaping, and surrounding buildings to redirect airflow and give maximum exposure to breezes. Use vegetation to funnel breezes and avoid wind dams that reduce the driving pressure differential around the building. Site objects should not obstruct inlet openings.
- The stack effect requires vertical distances between openings to take advantage of the effect; the greater the vertical distance, the greater the ventilation.
- Openings with areas much larger than calculated are sometimes desirable when anticipating increased occupancy or very hot weather.
- Horizontal windows are generally better than square or vertical windows. They produce more airflow over a wider range of wind directions and are most beneficial in locations where prevailing wind patterns shift.
- Window openings should be accessible to and operable by occupants.

- Vertical air shafts or open staircases can be used to increase and take advantage of stack effects. However, enclosed staircases intended for evacuation during a fire should not be used for ventilation.

Infiltration Measurement

Fan Pressurization

The fan pressurization method, sometimes called the "Minneapolis Blower Door," is used in measuring the amount of infiltration into the building and in locating leaks.

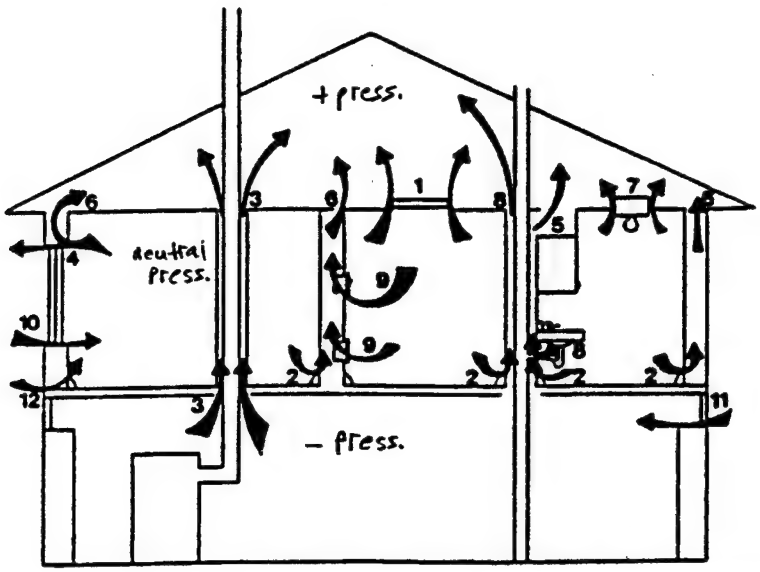
This method measures the building leakage rate independent of weather conditions. Equipment required for a quantitative measurement includes a blower (variable speed fan), a flow meter, a pressure gauge, and (optionally) a smoke source or an infrared scanning device to locate leaks. Also, a means of sealing the fan into the doorway is required so the only air going through the doorway passes through the fan.

The fan is generally used to move a large stream of air out of the building so that even the most minute streams of air (leaks) coming in may be detected. Moving air into or out of the building causes a different air pressure inside the building relative to the outside air pressure. If air is being forced out, the inside pressure is lower and vice versa.

When the inside pressure is low, air leaks into the building through any hole it can find in the exterior envelope of the structure. Leak locations can be found by checking suspected trouble spots for drafts with a smoke stick, an infrared camera, or even a person's hand.

Some of the common leak locations are shown in Figure D-5.

Figure D-5 also shows how air flows naturally through a building. As warm air rises, it tends to escape through cracks and holes near the top of the building. This escaping air causes a slight suction, which pulls in cold air through holes near the bottom of the building. These holes throughout the interior of the building need to be sealed to reduce air movement (heat loss).



Warm air escapes around:
1 attic hatch
2 baseboards and molding
3 chimney flue
4 doors and windows
5 dropped ceilings
6 exterior and partition walls
7 lighting fixtures
8 plumbing penetrations
9 wall switches and electrical outlets

Cold air infiltrates around:
10 doors and windows
11 rim joists
12 other cracks and holes

Figure D-5. Natural Air Flow Through a Building.

When calculating infiltration, it is important to select a method that facilitates the equipment needed to execute the method chosen. Any of the preceding methods may be applied in a variety of building types. If the blower door method is to be used, however, the portable air-tight door frame, variable speed fan, and pressure measuring gauges must be available.

Calculations by the air-change method have been compared to tracer gas measurements for two houses in California and two in Minnesota. The comparison showed that the air-change method can give estimates to within 20 percent of measurements for average construction under average conditions.

The air-change method should be used as a gross estimate at best. The accuracy of the crack method for design load calculations has restrictions of limited data on air leakage characteristics of components and by the difficulty of estimating pressure differences under appropriate design conditions of temperature and wind.

If a method is being selected for design purposes, some knowledge of infiltration will be necessary. In analyzing an existing condition, having the proper equipment and knowing how to use it, and knowing how to tabulate and analyze data will be required. To determine the degree of accuracy for the method chosen, the person(s) doing the analysis will need to determine their own degree of satisfaction in the selected method.

3 Ventilation

General Ventilation

General ventilation controls heat, odors, and contaminants. It may be provided by natural draft, by a combination of general supply and exhaust air fan and duct systems, by exhaust fans only (with replacement air through inlet louvers and doors), or by supply fans only (exhaust through relief louvers and doors).

It is important to provide at least a minimum amount of fresh air indoors, both for comfort and for health. Odors and a sense of staleness can be uncomfortable, and buildups of pollutants can be produced within buildings. These pollutants are easily removed with air changes through rooms.

Winter heat loss (and summer heat gain in closed, cooled buildings) occurs when fresh outdoor air enters a building to replace stale indoor air. This heat exchange must be calculated when sizing heating or cooling equipment or when estimating energy use per season.

Air exchange increases a building's thermal load in three ways. First, the incoming air must be heated or cooled from the outdoor air temperature to the indoor air temperature. Second, air exchange increases a building's moisture content, which means humid outdoor air must be dehumidified. Third, air exchange can increase a building's thermal load by decreasing the performance of the envelope insulation system. Air flowing around and through the insulation can increase heat transfer rates above design rates. Air flow within the insulation system can also decrease the system's performance due to moisture condensing in and on the insulation.

The calculation of the heat lost (or gained) by the introduction of outdoor air into spaces is:

$$q_v = (V) \times (1.08) \times (\Delta t)$$

where:

q_v = sensible heat exchange due to ventilation (Btu/h)

V = volume flow rate, in cubic feet per minute (cfm) of outdoor air introduced (see Q in Chapter 2 of this appendix and examples in infiltration section)

Δt = temperature difference between outdoor and indoor air °F

1.08 = A constant derived from the density of air at 0.075 $lb/cu\ ft$ under average conditions, multiplied by the specific heat of air (heat required to raise 1 lb of air 1 °F), which is 0.24 Btu/lb °F, and by 60 min/h. The units of this constant are Btu min/*cu ft* °F h.

Forced Ventilation

Fans can be used to forcibly introduce the desired amount of outdoor air directly into spaces. Fan manufacturers list their capacity in cubic feet per hour (cfh) or cubic feet per minute (cfm). This outdoor air can be blown into spaces, or it can be mixed with air being recirculated so that the different temperature of outdoor air is less noticeable.

Forced ventilation offers energy conservation opportunities if a heat exchanger is used. Outgoing and incoming airstreams can be kept separate but allow heat to transfer from one stream to the other. An example of this can occur during the winter months. Incoming very cold outdoor air can be given the heat, but not the pollutants, of outgoing warm indoor air. The reverse of this would happen in the summer.

To approximate the size of a fan, the following equations are used:

Q = (cfm outdoor air person) × (number of people)

or

Q = (cfm/sq ft floor area) × (sq ft floor area)

Q is the desired flow rate. The "cfm outdoor air person" and "cfm/sq ft floor area" expressions are ASHRAE's recommended design outdoor airflow rates. These values are found in Tables D-5 through D-7 on the following pages.

Estimating Heating/Ventilating Loads

The heating/ventilating loads of a building or an area of a building can be calculated from the following data:

1. Use required weather data tables to determine the outdoor design conditions.
2. Select the indoor design conditions for each room or space to be heated using the coldest weather. Determine each temperature difference (Δt).
3. Measure or estimate the temperatures in adjacent unheated spaces. Determine each temperature difference (Δt).
4. Calculate the net areas of all walls, glass, doors, ceilings, floors, partitions, etc., from building plans or from field measurements.
5. Determine the heat transmission loss coefficients ("U" values) for each area and type of construction. "U" is the thermal transmittance and is the overall expression of the steady state rate at which heat flows through architectural skin elements (walls, roofs, floors, etc.). This term is expressed in terms of Btu/h sq ft °F, and can be found from tables and charts found in the ASHRAE *Fundamentals Handbook*.

OUTDOOR AIR REQUIREMENTS FOR VENTILATION

COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements cfm/ person	L/s • person	cfm/ft^2	L/s • m^2	Comments
Dry Cleaners, Laundries						Dry-cleaning processes may require more air.
Commercial laundry	10	25	13			
Commercial dry cleaner	30	30	15			
Storage, pick up	30	35	18			
Coin-operated laundries	20	15	8			
Coin-operated dry cleaner	20	15	8			
Food and Beverage Service						
Dining rooms	70	20	10			
Cafeteria, fast food	100	20	10			
Bars, cocktail lounges	100	30	15			Supplementary smoke-removal equipment may be required.
Kitchens (cooking)	20	15	8			Makeup air for hood exhaust may require more ventilating air. The sum of the outdoor air and transfer air of acceptable quality from adjacent spaces shall be sufficient to provide an exhaust rate of not less than 1.5 cfm/ft^2 (7.5 L/s•m^2).
Garages, Repair, Service Stations						
Enclosed parking garage				1.50	7.5	Distribution among people must consider worker location and concentration of running engines; stands where engines are run must incorporate systems for positive engine exhaust withdrawal. Contaminant sensors may be used to control ventilation.
Auto repair rooms				1.50	7.5	
Hotels, Motels, Resorts, Dormitories				cfm/room	L/s • room	Independent of room size.
Bedrooms				30	15	
Living rooms				30	15	
Baths				35	18	Installed capacity for intermittent use.
Lobbies	30	15	8			
Conference rooms	50	20	10			
Assembly rooms	120	15	8			
Dormitory sleeping areas	20	15	8			See also food and beverage services, merchandising, barber and beauty shops, garages.
Gambling casinos	120	30	15			Supplementary smoke-removal equipment may be required.
Offices						
Office space	7	20	10			Some office equipment may require local exhaust.
Reception areas	60	15	8			
Telecommunication centers and data entry areas	60	20	10			
Conference rooms	50	20	10			Supplementary smoke-removal equipment may be required.
Public Spaces				cfm/ft^2	L/s • m^2	
Corridors and utilities				0.05	0.25	
Public restrooms, cfm/wc or cfm/urinal		50	25			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Locker and dressing rooms				0.5	2.5	
Smoking lounge	70	60	30			
Elevators				1.00	5.0	Normally supplied by transfer air.

* Table prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to control CO_2 and other contaminants with an adequate margin of safety and to account for health variations among people, varied activity levels, and a moderate amount of smoking.

**Net occupiable space.

Table D-5. Outdoor Air Requirements for Commercial Facilities.
Reprinted with permission from *ASHRAE Standard 62-1989.*

Application	Estimated Maximum** Occupancy P/1000 ft² or 100 m²	Outdoor Air Requirements cfm/person	L/s • person	cfm/ft²	L/s • m²	Comments
Retail Stores, Sales Floors, and Show Room Floors						
Basement and street	30			0.30	1.50	
Upper floors	20			0.20	1.00	
Storage rooms	15			0.15	0.75	
Dressing rooms				0.20	1.00	
Malls and arcades	20			0.20	1.00	
Shipping and receiving	10			0.15	0.75	
Warehouses	5			0.05	0.25	
Smoking lounge	70	60	30			Normally supplied by transfer air, local mechanical exhaust; exhaust with no recirculation recommended.
Specialty Shops						
Barber	25	15	8			
Beauty	25	25	13			
Reducing salons	20	15	8			
Florists	8	15	8			Ventilation to optimize plant growth may dictate
Clothiers, furniture				0.30	1.50	requirements.
Hardware, drugs, fabric	8	15	8			
Supermarkets	8	15	8			
Pet shops				1.00	5.00	
Sports and Amusement						
Spectator areas	150	15	8			When internal combustion engines are operated for maintenance of playing surfaces,
Game rooms	70	25	13			increased ventilation rates may be required.
Ice arenas (playing areas)				0.50	2.50	Incorporate systems for positive engine exhaust withdrawal.
Swimming pools (pool and deck area)				0.50	2.50	Higher values may be required for humidity control.
Playing floors (gymnasium)	30	20	10			
Ballrooms and discos	100	25	13			
Bowling alleys (seating areas)	70	25	13			
Theaters						
Ticket booths	60	20	10			Special ventilation will be needed
Lobbies	150	20	10			to eliminate special stage effects
Auditorium	150	15	8			(e.g., dry ice vapors, mists, etc.)
Stages, studios	70	15	8			
Transportation						
Waiting rooms	100	15	8			Ventilation within vehicles may require
Platforms	100	15	8			special considerations.
Vehicles	150	15	8			
Workrooms						
Meat processing	10	15	8			Spaces maintained at low temperatures
Photo studios	10	15	8			(-10°F to +50°F, or -23°C to +10°C) are not covered by
Darkrooms	10			0.50	2.50	these requirements unless the occupancy is continuous.
Pharmacy	20	15	8			Ventilation from adjoining spaces is permissible. When
Bank vaults	5	15	8			The occupancy is intermittent, infiltration will normally exceed the ventilation requirement.
Duplicating, printing				0.50	2.50	Installed equipment must incorporate positive exhaust and control (as required) or undesirable contaminants (toxic or otherwise).

INSTITUTIONAL FACILITIES

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements cfm/ person	L/s • person	cfm/ft^2	L/s • m^2	Comments
Education						
Classroom	50	15	8			
Laboratories	30	20	10			Special contaminant control systems may be required for processes or functions including laboratory animal occupancy.
Training shop	30	20	10			
Music rooms	50	15	8			
Libraries	20	15	8			
Locker rooms				0.50	2.50	
Corridors				0.10	0.50	
Auditoriums	150	15	8			
Smoking lounges	70	60	30			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Hospitals, Nursing, Convalescent Homes						
Patient rooms	10	25	13			Special requirements or codes and pressure relationships may determine minimum ventilation rates and filter efficiency. Procedures generating contaminants may require higher rates.
Medical procedure	20	15	8			
Operating rooms	20	30	15			
Recovery and ICU	20	15	8			
Autopsy rooms				0.50	2.50	Air shall not be recirculated into other spaces.
Physical Therapy	20	15	8			
Correctional Facilities						
Cells	20	20	10			
Dining halls	100	15	8			
Guard stations	40	15	8			

* Table prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to control CO_2 and other contaminants with an adequate margin of safety and to account for health variations among people, varied activity levels, and a moderate amount of smoking.

**Net occupiable space.

Table D-6. Outdoor Air Requirements for Institutional Facilities.
Reprinted with permission from *ASHRAE Standard 62-1989.*

6. Calculate the heat losses for walls, ceilings, partitions, glass, doors, and floors (above grade) to unheated areas using the equation:

$$Q = A \times U \times \Delta t \qquad \text{[Eq D-4]}$$

7. Calculate the heat losses for slab floors and basement walls below grade by using the equation:

$$Q = A \times U \times \Delta t$$

and a Dt determined by temperatures from tables and charts found in the ASHRAE *Fundamentals Handbook*.

OUTDOOR AIR REQUIREMENTS FOR VENTILATION OF RESIDENTIAL FACILITIES[a]
(Private Dwellings, Single, Multiple)

Applications	Outdoor Requirements	Comments
Living areas	0.35 air changes per hour but not less than 15 cfm (7.5 L/s) per person	For calculating the air changes per hour, the volume of the living spaces shall include all areas within the conditioned space. The ventilation is normally satisfied by infiltration and natural ventilation. Dwellings with tight enclosures may require supplemental ventilation supply for fuel-burning appliances, including fireplaces and mechanically exhausted appliances. Occupant loading shall be based on the number of bedrooms as follows: first bedroom, two persons; each additional bedroom, one person. Where higher occupant loadings are known, they shall be used.
Kitchens[b]	100 cfm (50 L/s) intermittent or 25 cfm (12 L/s) continuous or openable windows	Installed mechanical exhaust capacity[c]. Climatic conditions may affect choice of the ventilation system.
Baths, Toilets[b]	50 cfm (50 L/s) intermittent or 20 cfm (10 L/s) continuous or openable windows	Installed mechanical exhaust capacity[c].
Garages: Separate for each dwelling unit	100 cfm (50 L/s) per car	Normally satisfied by infiltration or natural ventilation.
Common for several units	1.5 cfm/ft^2 (7.5 L/s • m^2)	See "Enclosed parking garage," Table D-5

[a]In using this table, the outdoor air is assumed to be acceptable.
[b]Climatic conditions may affect choice of ventilation option chosen.
[c]Air exhausted from kitchens, bath, and toilet rooms may use air supplied through adjacent living areas to compensate for the air exhausted.

The air supplied shall meet the requirements of exhaust systems as described in 5.8 and be of sufficient quantities to meet the requirements of this table.

Table D-7. Recommended Outdoor Air Requirements for Residential Facilities.
Reprinted with permission from *ASHRAE Standard 62-1989.*

8. Calculate the heat losses for slab floors on grade by using the equation:

$$Q = F_2 \times P \times \Delta t \qquad \text{[Eq D-5]}$$

where P = perimeter of slab (feet), and F_2 is the "Heat Loss Coefficient of Slab Floor Construction" which can be found in the ASHRAE *Fundamentals Handbook.*

9. Calculate the infiltration of each room or area.

10. When outdoor air is introduced through a HVAC unit (makeup or ventilation air), that load must be part of the total ventilation load requirements when making the calculations. Use whichever load (ventilation or infiltration) is determined to be greater.

11. The total heating load is the sum of all the above heat transmission and infiltration/ventilation loads, which are considered to peak at the same time.

12. In buildings that have a permanent, steady internal heat source of considerable size (such as ovens, 24-h intensive lighting systems, etc.), an equivalent amount of heat could be deducted from the calculated total heating load, provided the load could not be cut off at a future date.

13. A limited amount of additional heating capacity should be added for a "pick-up load" for buildings that have night and/or weekend setback or are intermittently heated.

Design Considerations

General ventilation may be provided with either natural or mechanical supply and/or exhaust systems. Some ventilation systems must handle simultaneous exposures to hazardous substances and heat. In such cases, ventilation may consist of a combination of local, general supply, and exhaust air systems. Some factors to consider in selection and design are as follows:

- Local exhaust systems provide general ventilation for the work area.
- A balance of the supply and exhaust systems is required for either system to function as designed.
- Natural ventilation systems are most applicable when internal heat loads are high, and the building is tall enough to produce a significant stack effect.
- To provide effective general ventilation for heat relief by either natural or mechanical supply, the air must be delivered in the work zones (no more than 10 ft above the floor) with an appreciable air velocity. A sufficient exhaust volume is necessary to remove the heat liberated in the space. Local relief systems may require supplementary supply air for heat removal.
- Supply and exhaust air cannot be used interchangeably. Supply air can be delivered where it is wanted at controlled velocities, temperature, and humidity. Exhaust systems should be used to capture heat and fumes at the source.
- General building exhaust may be required in addition to local exhaust systems.
- The exhaust discharge, whether local or general, should be located where it will not be recirculated.

Ventilation Air Velocity

The level of air motion at the worker is important. At fixed work positions with light activity, the velocity should not exceed 200 fpm for continuous exposure. With high work levels and intermittent exposures, velocities of 400 to 800 fpm may be used. When high-velocity air is used, it is important to avoid the undesirable effects of hot air convection and disturbance of local exhaust ventilation systems. Table D-8 lists some acceptable air motion rates.

Exposure	Air Velocity, fpm
Continuous	
Air-conditioned space	50 to 75
Fixed workstation, general ventilation, or spot cooling	
Sitting	75 to 125
Standing	100 to 200
Intermittent, spot cooling, or relief stations	
Light heat loads and activity	1000 to 2000
Moderate heat loads and activity	2000 to 3000
High heat loads and activity	3000 to 4000

Table D-8. Acceptable Air Motion at the Worker.

Locker Room, Toilet, and Shower Space Ventilation

The ventilation of locker rooms, toilets, and shower spaces is important in removing odor and in reducing humidity. State and local regulations should be consulted when designing these facilities.

Supply air may be introduced through door or wall grilles. In some cases, plant air may be so contaminated that filtration or mechanical ventilation may be required. When control of workroom contaminants is inadequate, the total exposure to employees can be reduced by making sure that the level of contamination in the locker rooms, lunchrooms, and break rooms is minimized by pressurizing these areas with excess supply air.

When mechanical ventilation is used, the supply system should have supply fixtures such as wall grilles, ceiling diffusers, or supply plenums to distribute the air adequately throughout the area.

In locker rooms, the exhaust should be taken primarily from the toilet and shower spaces, as needed, and the remainder from the lockers and the room ceiling. Table D-9 provides a guide for ventilation of these spaces.

Description	Inch-Pound Units	SI Units
Locker Rooms		
Coat hanging or clean change room for nonlaboring shift employees with clean work clothes	1 cfm/sq ft	5 L/s•m^2
Change room for laboring employees with wet or sweaty clothes	2 cfm/sq ft; 7 cfm exhausted from each locker	10 L/s•m^2; 3 L/s exhausted from each locker
Change room for heavy laborers or workers assigned to working and cleaning where clothes will be wet or pick up odors	3 cfm/sq ft; 10 cfm exhausted from each locker	20 L/s•m^2; 5 L/s exhausted from each locker
Toilet Spaces	2 cfm/sq ft; at least 25 cfm per toilet facility; 200 cfm min.	10 L/s•m^2; at least 10 L/s per toilet facility; 90 L/s min.
Shower Spaces	2 cfm/sq ft; at least 50 cfm per shower head; 200 cfm min.	10 L/s•m^2; at least 20 L/s per shower head; 90 L/s min.

Table D-9. Ventilation for Locker Rooms, Toilets, and Shower Spaces.

4 Exhaust

Exhaust ventilation systems collect and remove airborne contaminants consisting of dusts, fumes, mists, fibers, vapors, and gases that can create an unsafe, unhealthy, or undesirable atmosphere. Exhaust systems are used to remove impurities from the air at the source, preventing them from contaminating the bulk of the air in the building. Necessary air changes are then held to a minimum.

Replacement air, which is usually conditioned, provides air to the work space to replace exhausted air, and the systems are not isolated from each other. A complete industrial ventilation program includes replacement air systems that provide a total volumetric flow rate equal to the total exhaust rate. If insufficient replacement air is provided, the pressure of the building will be negative relative to local atmospheric pressure. Negative pressure allows air to infiltrate through open doors, window cracks, and combustion equipment vents.

There are two types of exhaust systems:

1. General Exhaust, in which an entire work space is exhausted without considering specific operations.
2. Local Exhaust, in which the contaminant is controlled at its source.

General Exhaust/Dilution Ventilation

The terms "general exhaust" and "dilution ventilation" are often used interchangeably. This type of exhaust refers to dilution of contaminated air with uncontaminated air in a general area, room, or building for the purpose of health hazard or nuisance control.

In general, dilution ventilation is not as satisfactory for health hazard control as is local exhaust. In some cases, dilution ventilation must be used because the operation or process prohibits local exhaust. Circumstances may be found in which dilution ventilation provides an adequate amount of control more economically than a local exhaust system. Economical considerations should not be

based entirely upon the first cost of the system because dilution ventilation frequently exhausts large volumes of heat from a building and can easily be a troublesome factor.

The use of dilution ventilation has four limiting factors:

1. The quantity of contaminant generated must not be too great or the air volume necessary for dilution will be impractical.
2. Workers must be far enough away from contaminant evolution, or evolution of contaminant must be in sufficiently low concentrations so workers will not have an exposure in excess of the established Threshold Limit Values (TLVs).
3. The toxicity of the contaminant must be low.
4. The evolution of contaminants must be reasonably uniform.

Dilution ventilation is seldom applied to fumes and dusts because the high toxicities often encountered require too great a quantity of dilution air, velocity and rate of evolution are usually very high, and data on the amount of fumes and dust production are very difficult, if not impossible, to obtain.

Dilution ventilation is most often used to advantage to control the vapors from organic liquids such as the less toxic solvents. To successfully apply the principles of dilution to such a problem, factual data are needed on the rate of vapor generation or on the rate of liquid evaporation.

Basic Principles

Some basic principles to be applied to a dilution ventilation system are as follows:

- From factual data, select the amount of air required for satisfactory dilution of the contaminant.

- Locate the exhaust openings near the sources of contaminant if possible, in order to obtain the benefit of spot ventilation.

- For dilution methods to be effective, the exhaust outlet and air supply must be located so that all the air used in the ventilation passes through the zone of contamination.

- Replace exhausted air by a make-up air system. Make-up air should be heated during cold weather. Dilution ventilation systems usually handle large quantities of air by means of propeller fans. Make-up air usually must be provided if the ventilation is to be adequate and the system is to operate satisfactorily.

- The general air movements in the room caused by suction at the exhaust opening should keep the contaminated air between the operator and the exhaust opening, and not draw contaminants across the operator.

- A combined supply and exhaust system is preferred with a slight excess of exhaust if there are adjoining occupied spaces, and a slight excess of supply if there are no such spaces.

- Avoid re-entrance of the exhausted air by discharging the exhaust high above the roof line, or by assuring that no window, outside air intakes, or other such openings are near the exhaust discharge.

Local Exhaust

Local exhaust is preferable because it offers better contaminant control with minimum air volumes. This, in turn, lowers the cost of air cleaning and replacement air equipment.

Local exhaust systems can be classified as: (1) Constant Air Volume or (2) Variable Air Volume, based on the method of system operation and control. Each of these classifications can be further broken down into individual or central systems based on the arrangement of the major system components such as the fans, plenums, or duct mains and branches.

Constant Air Volume Systems

This type of system exhausts a fixed quantity of air from each safety cabinet, fume hood, or room module. Constant air volume systems will handle the same exhaust air quantity for any condition. For this reason, the capacities of the exhaust air and supply air systems will limit the total number of fume hoods and room modules to be installed. This type of system is flexible with respect to location of hoods but may incur high ownership and operating costs because of the

large air volumes handled. These high costs may impose a limitation on the total number of hoods or modules that can be installed in the building.

Constant air volume systems are highly stable in operation and simple to balance. In most installations, there is no need for continuous adjustment of air balance during normal operation.

Variable Air Volume Systems

Variable air volume systems can shut down inactive fume hoods and room modules. This capability results in an economic system that reduces the air flow during periods when some of the hoods and room modules are not in use, and the exhaust air system is operated at less than full capacity. More freedom in the installation of the hoods and room modules is possible since the total number of units that may be connected does not entirely depend on the capacity of the exhaust system.

Variable air volume systems are not as stable in operation as constant air volume systems are. They are also more difficult to balance and control. Sensitive instrumentation and controls are required, which result in high initial and maintenance costs. Reliability in a corrosive atmosphere is highly questionable. For some applications, the use of balancing dampers in exhaust air ducts is prohibited by codes.

One problem associated with the variable air volume system is the regulation of the total simultaneous operating usage to match design usage factors. If the collective area of operating hood openings at any one time exceeds design opening diversity values, the proper face velocity requirements will not be achieved and personnel could be endangered. Visual and audible alarms should be equipped on hoods to warn workers of unsafe air flows.

Individual exhaust air systems. Individual exhaust air systems use a separate exhaust air fan, exhaust connection, and discharge duct for each hood or module. The exhaust for the hood or module served by the individual exhaust system does not directly affect the operation of any other area of the building, which permits selective operation of individual hoods and modules by starting or stopping the fan motor.

The recommended operation is for exhaust air fans to be on at all times and to be electrically interlocked so that, if any critical exhaust air fan is shut down, the

supply air fans will shut down. Although more fans are used than for central systems, the overall space requirements are usually less for individual systems because of the small, direct duct connection. The use of more fans does increase capital and maintenance costs.

The shutdown of individual exhaust air systems will upset the proper directional air flow and may cause potentially hazardous contaminants and odors to flow into the corridor and adjacent rooms. If this type of system is used, precautions to reverse air flow (such as air locks) should be provided.

Central exhaust air systems. Central exhaust air systems consist of a common suction plenum, one fan, and branch connections to multiple exhaust terminals. This type of system generally costs less than individual exhaust air systems, costs less to maintain, permits low cost standby exhaust air fan provisions, and is applicable to remote high stack discharge requirements. Central systems are more difficult to balance and may have difficulties with parallel fan operation. The central exhaust air system is best when exhausting similar types of units such as laboratory fume hoods.

5 Fans

The fan is an air pump that causes airflow by creating a pressure difference. Fans produce pressure and/or flow by rotating blades of the impeller, imparting kinetic energy to the air by changing its velocity. By definition, the term "fan" is limited to devices producing pressure differentials of less than 28 in. w.g. at sea level. The following definitions and equations will help in the understanding of fans and their function in a system.

- *Brake Horsepower*—The actual horsepower required to drive the fan. This number is greater than a theoretical "air horsepower" because it includes loss due to turbulence and other inefficiencies in the fan, plus bearing losses. It is the power furnished by the fan motor.

- *Fan Air Volume*—The cubic feet per minute (cfm) of air handled by a fan at any air density. This is different from the cubic feet per minute of standard air (scfm), which is at 0.075 lb/ft.

- *Fan Outlet Velocity*—The theoretical velocity of the air as it leaves the fan outlet. This velocity is calculated by dividing the air volume in cfm by the fan outlet area in square feet. Since the velocity varies over the cross-section of all fan outlets, this value is only a theoretical value that could occur at a point removed from the fan. Because of this, all velocity readings, including total pressure and static pressure, should be taken farther along in a straight duct connected to the fan discharge where the flow is more uniform.

- *Fan Static Pressure (SP)*—The fan total pressure (TP) less the fan velocity pressure (VP) as shown in Figure D-6.

$$SP = TP_{(outlet)} - TP_{(inlet)} - VP_{(outlet)}$$

$$VP_{(outlet)} = TP_{(outlet)} - SP_{(outlet)}$$

$$SP = SP_{(outlet)} - TP_{(inlet)}$$

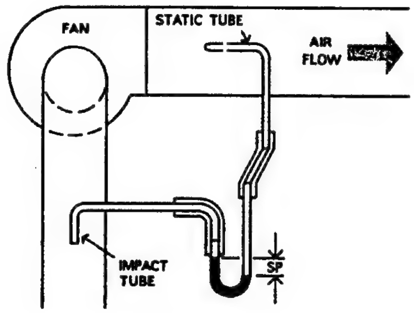


Figure D-6. Fan Static Pressure.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

- *Fan Total Pressure*—The difference between the total pressure at the fan outlet and the total pressure at the fan inlet. This value measures the total mechanical energy added to the air or gas by the fan (Figure D-7).

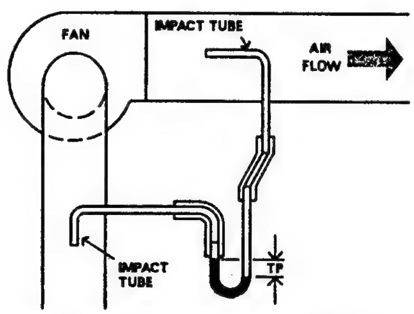


Figure D-7. Fan Total Pressure.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

- *Fan Velocity Pressure*—The pressure corresponding to the fan outlet velocity. It is the measure of kinetic energy per unit volume of flowing air (Figure D-8).

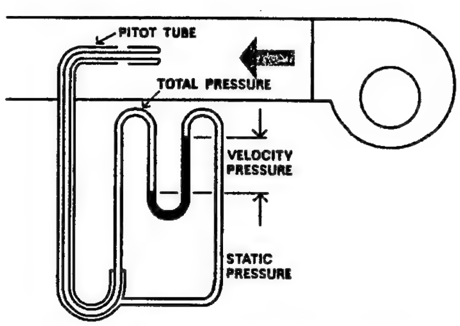


Figure D-8. Fan Velocity Pressure.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

- *Tip Speed (TS)*—The circumference of the fan wheel times the rpm of the fan, expressed in ft/min. Also known as peripheral velocity (Figure D-9).

$$TS = \frac{D \times rpm}{12}$$

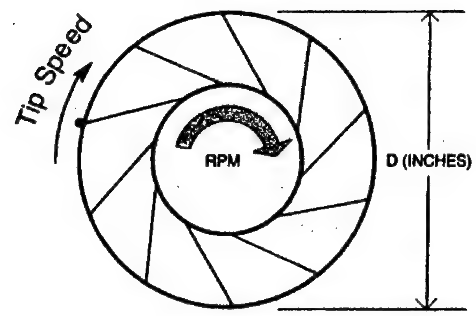


Figure D-9. Tip Speed.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Fan Types

Fan types are generally classified by the direction of air flow through the impeller. The two main types are:

1. Centrifugal
2. Axial.

Centrifugal Fans

Centrifugal fans consist of a number of blades that are inclined in a direction opposite to the fan rotation, in a vertical position (Figure D-10). This arrangement enables the wheel to operate at a lower tip speed, giving more cfm at a lower rpm at a given static pressure. Nonoverloading characteristics are also associated with this type of fan. Centrifugal fans are used the most in comfort applications because of its wide range of quiet, efficient operation at comparatively high pressures. The centrifugal fan inlet can be readily attached to an apparatus of large cross-section, while the discharge is easily connected to relatively small ducts. Air flow can be varied to match air distribution system requirements by simple adjustments to the fan drive or control devices.

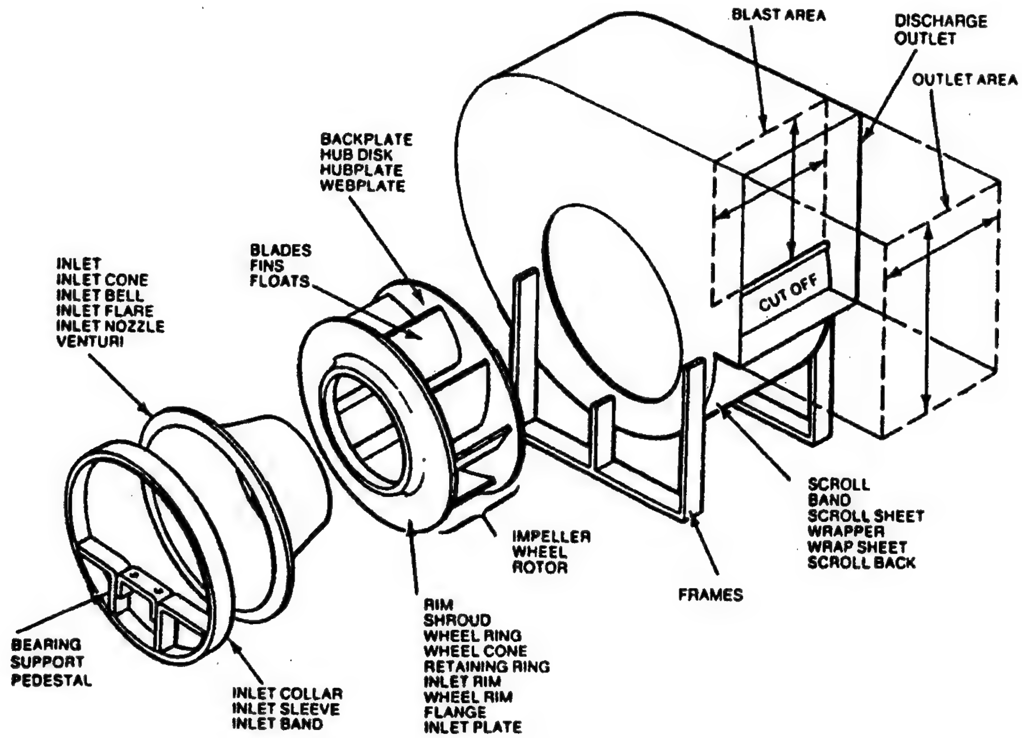


Figure D-10. Centrifugal Fan Components.

Reprinted from AMCA Publication 201-90 with written permission from the Air Movement and Control Association International, Inc.

Centrifugal fan types include:

1. Airfoil
2. Backward-curved blade
3. Radial blade
4. Forward-curved blade.

Examples of these fan types are shown in Figures D-11 through D-14. The figures are based on Air Movement and Control Association International, Inc. (AMCA) Publication 201-90.

Airfoil fans. Airfoil blades curve away from the direction of rotation, and fans consist of 10 to 16 blades. Relatively deep blades provide efficient expansion within the blade passages. When this blade is properly designed, this will be the most efficient and the highest speed of the centrifugal fan designs. The static efficiency of these fans is around 86 percent. The clearance and alignment between the wheel and inlet bell need to be very close to reach the maximum efficiency capacity. A scroll-type housing is usually used (Figure D-11).

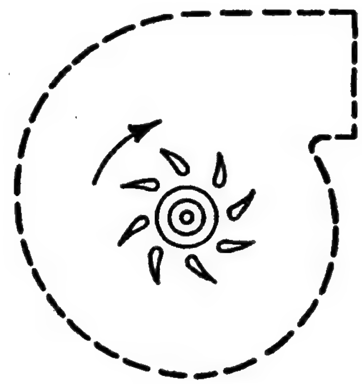


Figure D-11. Airfoil Fan.

Backward-curved fans. Fan blades are inclined in a direction opposite to the fan rotation (Figure D-12). Blades are single thickness, and fans consist of 10 to 16 blades. Fan efficiency is slightly lower than that of the airfoil fan. These fans travel at about twice the speed of the forward-curved fans. Normal selection range is usually 40 to 80 percent of wide open air flow. The static pressure proportion of the total pressure discharge is 70 percent, while the velocity pressure is 30 percent. For a given selection, the larger the fan, the more efficient it will be. Some advantages of the backward-curved fan are higher efficiency and nonoverloading characteristics. This type of blade allows material buildup and should only be used on clean air containing no condensable fumes or vapors. It is normally used for high capacity, high pressure applications where power savings may outweigh its higher first cost. Larger shaft and bearing sizes are required for higher speeds. Because of this, proper wheel balance is more important. Housing designs closely resemble those of the airfoil designs.

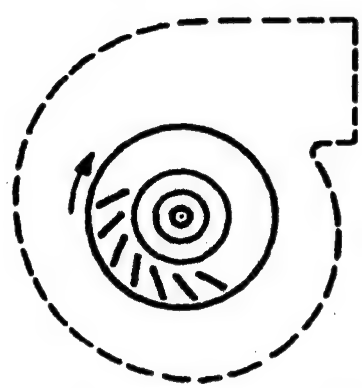


Figure D-12. Backward-Curved Fan.

Radial blade fans. Radial blade fans (Figure D-13) are used for systems handling materials likely to clog the fan wheel. These fans usually have medium tip speed and noise factor and are used for buffing exhaust, woodworking exhaust,

or for applications where a heavy dust load passes through the fan. This type of blade is the simplest of all centrifugal fans and the least efficient. Horsepower rises with increasing air quantity in an almost directly proportional relation, which can lead to overloading. Fans usually include 6 to 10 blades, and the wheel is easily repaired. A scroll-type housing is usually used.

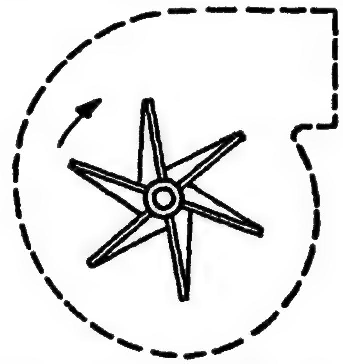


Figure D-13. Radial Blade Fan.

Forward-curved fans. The leading edges of these blades curve toward the direction of rotation. These fans usually consist of 24 to 64 shallow blades that have both the tip and heel curved forward. The efficiency of this fan is somewhat less than airfoil and backward-curved fans. Lightweight and low-cost construction, low space requirements, low tip speeds, and quiet operation are some common characteristics. Air leaves the wheel at a velocity greater than the wheel tip speed, and the primary energy is transferred to the air by use of high velocity in the wheel. The slow speed of this fan minimizes the shaft and bearing size, and it has a wide operating range, from 30 to 80 percent wide open volume. The static pressure proportion of the total pressure discharge is 20 percent, while the velocity pressure is 80 percent. Horsepower increases continuously with increasing air quantity. These fans are not recommended for fumes or dusts that would stick to the short curved blades because they would cause unbalance and would make cleaning difficult. These fans are typically used for producing high volumes at low static pressure. A scroll-type housing is usually used.

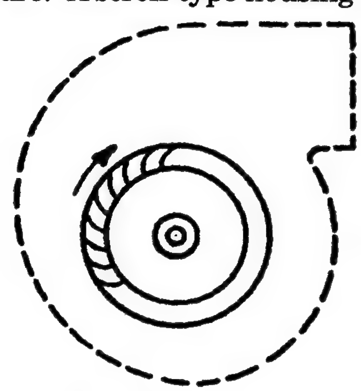


Figure D-14. Forward-Curved Fan.

Axial Fans

Axial flow fans consist of two or more blades. They are used for moving large quantities of air against a lower static pressure, and their common usage is for general ventilation. Pressure is produced from the change in velocity passing through the impeller, with none being produced by centrifugal force. Axial fan blades are divided into three types:

1. Propeller
2. Tubeaxial
3. Vaneaxial.

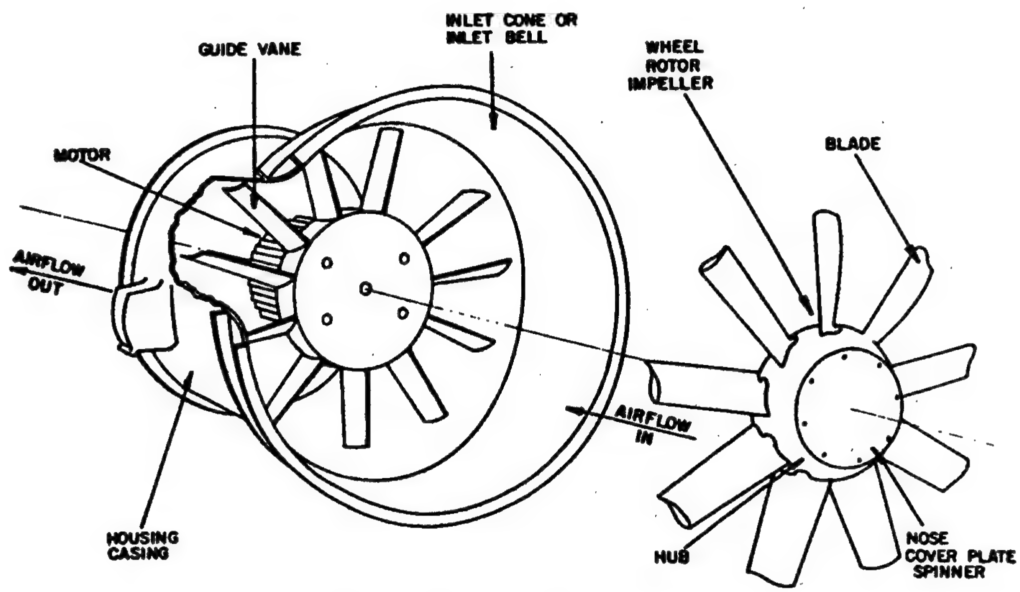
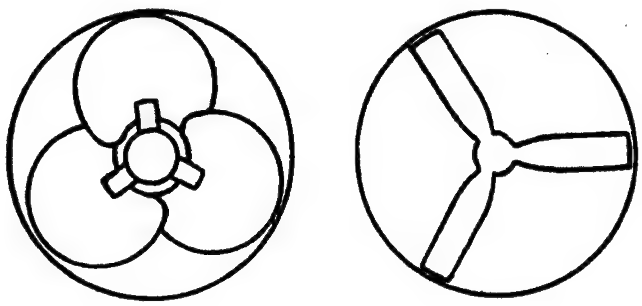


Figure D-15. Axial Fan Components.

Reprinted from AMCA Publication 201-90 with written permission from the Air Movement and Control Association International, Inc.

Propeller fans. Propeller fans are usually of inexpensive construction. Impellers are made of two or more blades, which are generally of single thickness and are attached to smaller hubs. Velocity pressure is the primary form of energy transfer. These fans work well in transferring high volumes of air at little or no static pressure differential. Propeller fans have the lowest efficiency of axial fans. Disc-propeller fans are used for moving clean air against no duct resistance. Housing generally consists of a circular ring, with the best performance coming from designs in which the housing is close to the blade tips (Figure D-16).



(a) Disc Blade (b) Propeller Blade

Figure D-16. Propeller Fans.

Environmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

Tubeaxial fans. Tubeaxial fans (Figure D-17) consist of 4 to 8 blades, with the hub usually less than 50 percent of fan tip diameter. This type of fan is somewhat more efficient than the propeller fan design. It is best suited to moving air containing condensable fumes, pigments, and other materials that will collect on fan blades. Housing is a cylindrical type that has a close clearance between the tube and wheel tip.

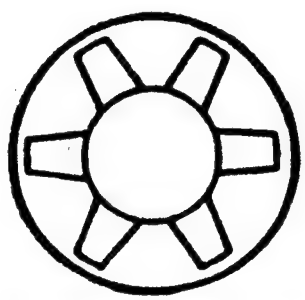


Figure D-17. Tubeaxial Fan.

Vaneaxial fans. Vaneaxial blades are adjustable, fixed, or controllable pitch types with the hub usually being greater than 50 percent of fan tip diameter. Vaneaxial fans (Figure D-18) have the highest efficiency of axial fans and can reach higher pressures. The operating range (cfm per fan) for axial fans is from 65 to 90 percent. The most efficient vaneaxial fans are those with airfoil blades, which should only be used with clean air. Vaneaxial fans are generally used for handling large volumes of air at low static pressures. Housing consists of a cylindrical tube that fits closely to the outer diameter of the blade tips.

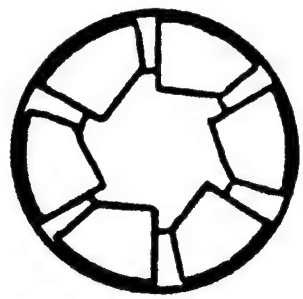


Figure D-18. Vaneaxial Fan.

Special Fan Types

Inline flow centrifugal fans. This type of fan (Figure D-19) has backward-curved blades and a special housing that permits a space-saving straight-line duct installation. The wheel is very similar to that of the airfoil. Space requirements are similar to a vaneaxial fan.

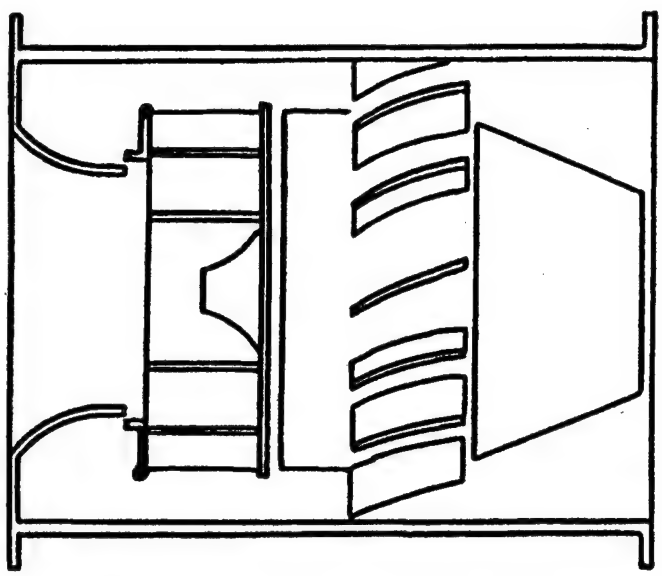
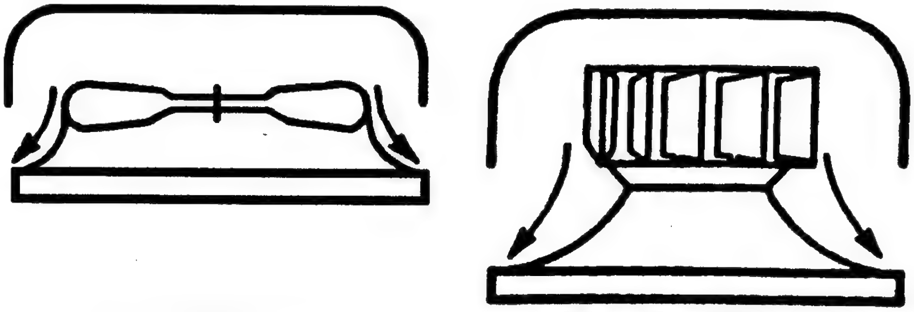


Figure D-19. Inline Flow Centrifugal Fan.

Power roof ventilators. The objective of these ventilators (Figure D-20) is to produce a high-volume flow rate at low pressure. They can be of centrifugal fan type or axial fan type.



(a) Axial Ventilator (b) Centrifugal Ventilator

Figure D-20. Power Roof Ventilators.

Adapted with permission from ASHRAE *1988 Equipment*.

Fan Classifications

Fan classifications are based on fan speeds and static pressures. "Class" refers to an AMCA standard that was developed to reflect operating conditions of the impellers, bearings, and housing of fans. Fan classifications vary with respect to impeller design type and other criteria. Figure D-21 shows an example of fan classifications.

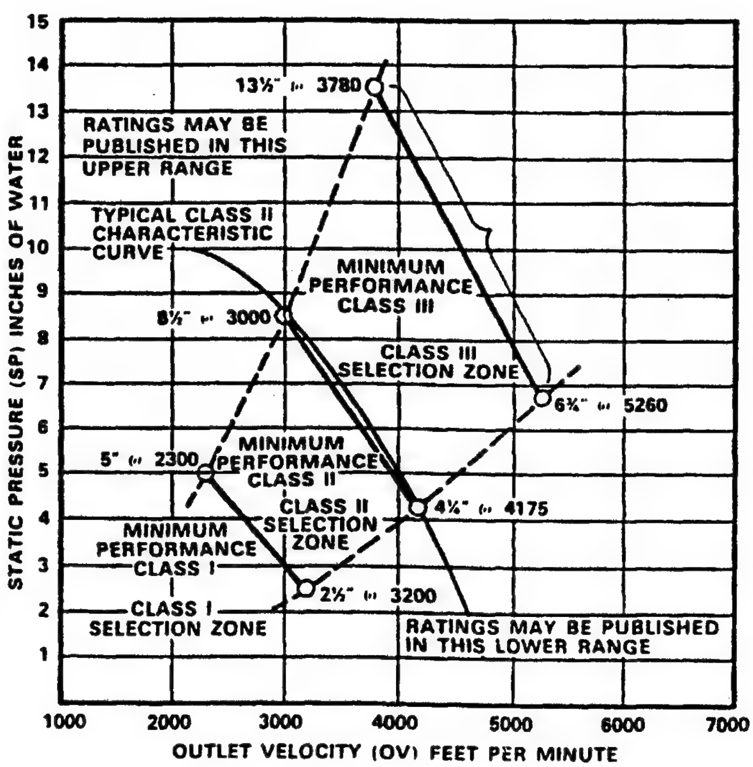


Figure D-21. Fan Class Standards.

Reprinted from AMCA 99-2408-69 with written permission from the Air Movement and Control Association International, Inc.

The published ratings of a fan should always be checked to make sure that revised operating conditions do not require a different class of fan. This type of change could also change the pressure classification of part or all of a duct system.

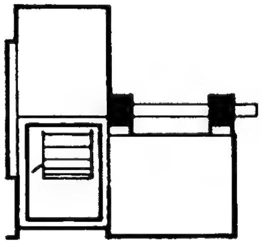
Fan Drives

Fan drive arrangements, which are standardized by AMCA, refer to the relation of the fan wheel to the bearings and the number of fan inlets. The fan drive may be belt or direct driven. First cost and space requirements are the major factors when selecting a suitable fan arrangement. A single inlet fan is about 30 percent taller than the double inlet type but only about 70 percent as wide for the same capacity. Double inlet fans are lower in cost in the larger sizes, while single inlet fans are usually less expensive in the smaller sizes. Drive arrangements designated by AMCA for centrifugal fans are shown in Figure D-22. Drive arrangements designated by AMCA for in-line fans are shown in Figure D-23.

Different motor locations for a belt-driven fan are shown in Figure D-24. This location is always determined by facing the drive side of the fan or blower and is independent of the discharge or rotation. Positions W and Z have the simplest fan base and belt guard construction.

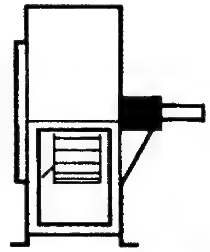
Rotation, clockwise or counter clockwise, is determined by the direction the fan wheel will be turning as viewed from the drive side of the fan (Figure D-25). The drive side of a single inlet fan is considered to be the side opposite the inlet, regardless of the actual drive location. When fans are to be inverted for ceiling suspension, the direction is determined when the fan is resting on the floor.

Most fans are driven at constant speed by constant speed motors and commonly deliver a constant air quality. Motors range from single phase and small fractional horsepower to large polyphase motors. The installed motor should be checked for sufficient starting torque to overcome the inertia of the fan wheel and drive package, and accelerate the fan to its design speed. A "V" belt is usually used to connect the motor to the driven fan. This belt also allows the synchronous speed of the motor to be converted to a lower, proper speed of the fan.



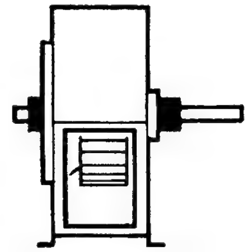
Arrangement 1 SWSI

For belt drive or direct connection, impeller overhung, two bearings on base.



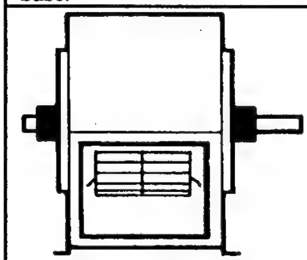
Arrangement 2 SWSI

For belt drive or direct connection, impeller overhung, bearings in bracket supported by fan housing.

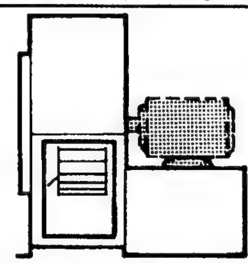


Arrangement 3 SWSI

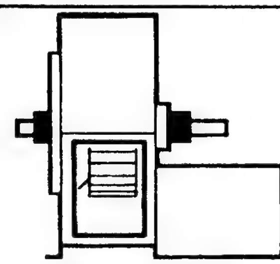
For belt drive or direct connection, one hearing on each side and supported by fan housing.



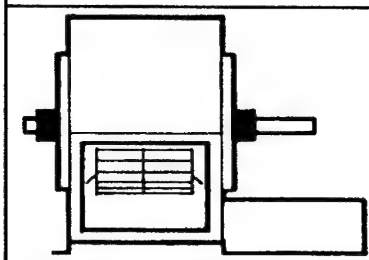
Arrangement 3 DWDI
For belt drive or direct connection, one hearing on each side and supported by fan housing.



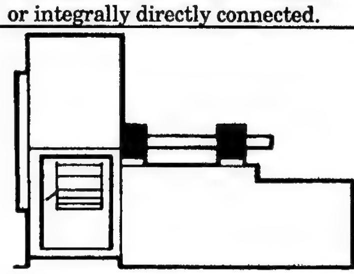
Arrangement 4 SWSI
For direct drive, impeller overhung on prime mover shaft. No hearings on fan. Prime mover base mounted or integrally directly connected.



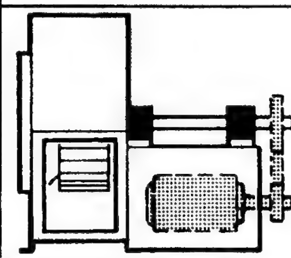
Arrangement 7 SWSI
For belt drive or direct connection. Arrangement 3 plus base for prime mover.



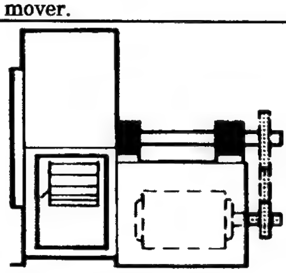
Arrangement 7 DWDI
For belt drive or direct connection, arrangement 3 plus base for prime mover.



Arrangement 8 SWSI
For belt drive or direct connection, arrangement 1 plus extended base for prime mover.



Arrangement 9 SWSI
For belt drive, impeller overhung, two bearings with prime mover outside base.

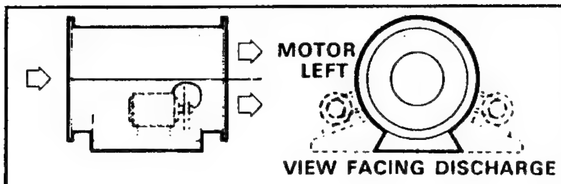


Arrangement 10 SWSI
For belt drive, impeller overhung. two bearing with prime mover inside base.

SW - Single Width DW - Double Width SI - Single Inlet DI - Double Inlet

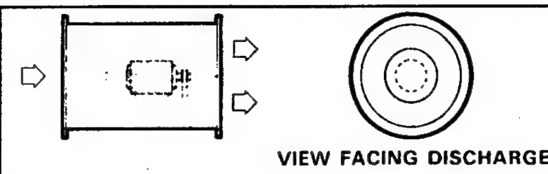
Figure D-22. Drive Arrangements for Centrifugal Fans.

Reprinted from AMCA 99-2408-69 with written permission from the Air Movement and Control Association International, Inc.



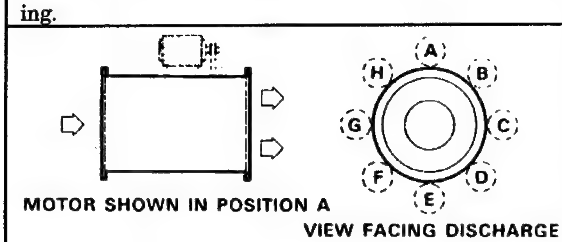
Arrangement 1

Belt drive with motor mounted independent of fan casing--typically used for motors too large for fan casing.



Arrangement 4

Direct drive with wheel overhung on motor shaft, can be used with vertical units.



Arrangement 9

Belt drive with motor located on periphery of casing in one of eight standard locations, can be used with vertical units.

Figure D-23. In-Line Fans.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

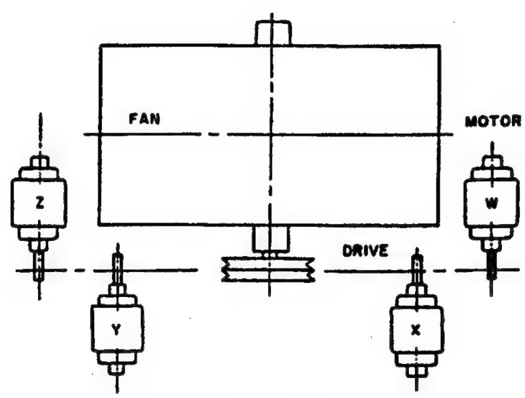


Figure D-24. Motor Positions.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

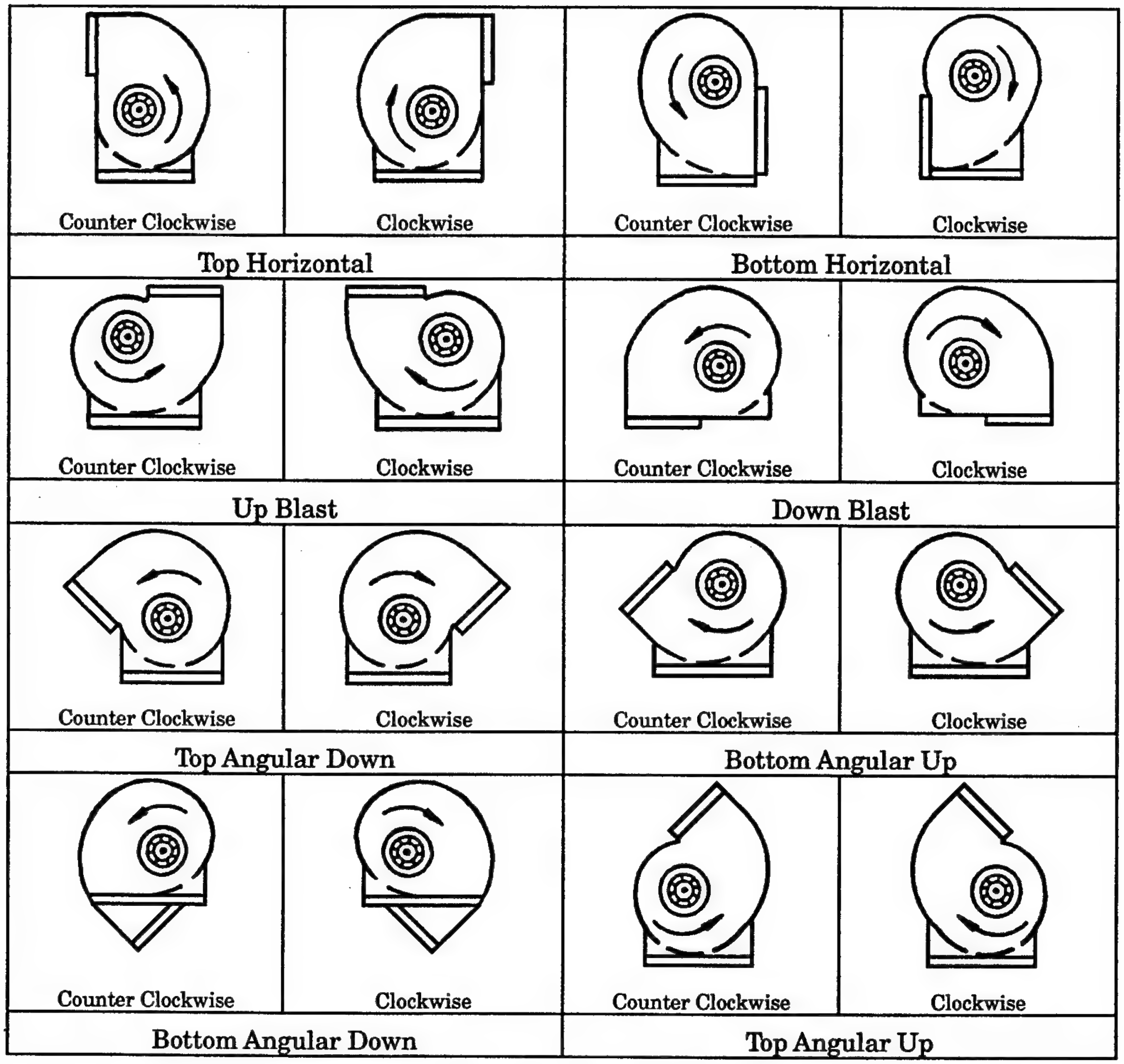


Figure D-25. Direction of Rotation and Discharge.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Some common conditions to consider in designing a satisfactory drive are:

1. Drives should be installed with provisions for center distance adjustment. This provision is important because all belts stretch.
2. Centers should not exceed 2-1/2 to 3 times the sum of the sheave diameters or be less than the diameter of the larger sheave.
3. The arc of contact on the smaller sheave should not be less than 120 degrees.
4. Sheave diameter ratios should not exceed 8:1.
5. Belt speed preferably should not exceed 5,000 ft/min, or be less than 1,000 ft/min—4,000 ft/min is the best practice.

6. Sheaves should be dynamically balanced for speeds in excess of 5,000 ft/min rim speed.

Some helpful points to watch for when installing drives are as follows:

1. Be sure that shafts are parallel and sheaves are in proper alignment. Check again after a few hours of operation.
2. Do not drive sheaves on or off shafts. Wipe shaft, key, and bore clean with oil. Tighten screws carefully. Recheck and retighten after a few hours of operation.
3. Belts should never be forced over sheaves.
4. In mounting belts, be sure the slack in each belt is on the same side of the drive. This side should be the slack side of the drive.
5. Belt tension should be reasonable. When in operation, the tight side of the belts should be in a straight line from sheave to sheave, and with a slight bow on the slack side. All drives should be inspected periodically to be sure belts are under proper tension and are not slipping.
6. When making replacements of multiple belts on a drive, be sure to replace the entire set with a new set of matched belts.

Fan Noise

One major cause of fan noise is surge. This is the result of periodic vibrations of the fan and ducts connected to it. It is caused by unstable operation of the fan. Surge commonly occurs when the actual static pressure is high, compared to the static pressure that the fan can reach at the particular speed at which it is operating. One way to check for surge is to relieve fan static pressure.

Another cause of fan noise is resonance. This will result in one or more sections of the duct system vibrating at the same frequency as a vibration produced by the fan. This can be checked by changing the fan speed by ±10 percent and noting whether the vibration stops.

If the fan performance is not matched to the duct system, fan noise will increase. One possibility is that the fan may be handling more air than required. Reducing the fan speed would reduce noise.

Air flow at the entrance and exit of a fan should be as smooth as possible to minimize the generation of turbulence. Conditions that produce turbulent air

flow usually result in greater noise generation and increase static pressure drop in the system. The air flow on the outlet side of a fan is always turbulent for at least 3 to 6 duct diameters downstream. Fittings (such as elbows or sudden transitions) placed closer to the fan than this distance may result in noise problems. Figure D-26 shows some examples of good and bad fan outlet conditions.

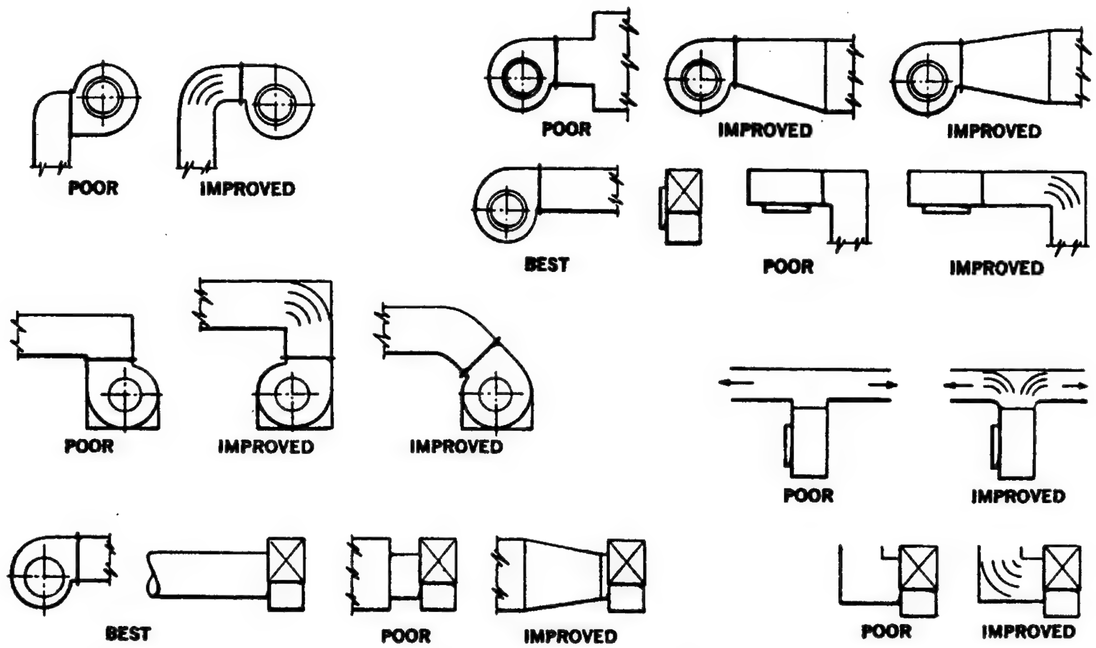


Figure D-26. Fan Outlet Conditions.

SMACNA, *HVAC Systems - Testing, Adjusting & Balance*, 2nd Ed., 1993. Used with permission.

Flexible connectors should be used on fans at each duct connection (Figure D-27). These connectors should not be pulled taut, but should be long enough to provide folds or flexibility when the fan is off.

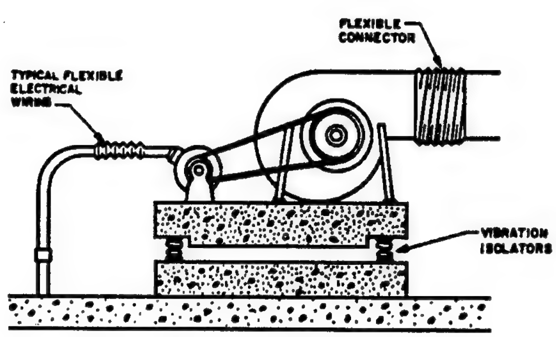


Figure D-27. Flexible Connections.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Steel springs are usually used as noise isolators on fans to absorb their vibrations. Concrete bases are also used with fans because they can reduce the

amplitude of oscillation of the equipment. This reduction, in turn, reduces the transmission of vibration through connected piping and duct.

Fan Selection

The following information is necessary before proceeding with any fan selection:

1. *Volume Required*—The first step to consider in selecting a fan or ventilator is the total cfm of the space. This can be accomplished in two different ways.

   a. *Air change method*—This method involves calculating the required number of air changes necessary to give the proper ventilation for a given space, and the total cubic feet of air space of the building. Air changes required must conform to the local health department code covering the type of installation. If no local codes are to be met, tables of air changes can be used.

   $$cfm = \frac{building\ volume\ in\ cubic\ feet}{\min/air\ change} \qquad \text{[Eq D-6]}$$

   b. *Heat removal method*—The average outside temperature, desired inside temperature, and Btu per minute are required to use this method. This formula gives the amount of air to be passed through the building to maintain desired inside temperature. The cfm deals primarily with heat, which changes the temperature of the substance involved, or sensible heat. It can be applied to installations where any general ventilation of a heat problem is desired.

   $$cfm = \frac{total\ Btu\ per\ \min}{0.018 \times temperature\ rise\ {}^\circ F} \qquad \text{[Eq D-7]}$$

2. *Fan Static Pressure*—The fan total pressure less the fan velocity pressure.
3. *Type of Material Handled Through Fan*—These include explosive fumes, general ventilation, fibrous material (heavy dust load), removal of heat, or corrosive fumes.
4. *Direct or Belt Driven*—Direct-driven exhausts offer a more compact assembly, and assure constant fan speed. They eliminate belt slippage that occurs when belt-driven drives are not maintained. Fan speeds are limited to available motor speeds. Belt-driven drives are often preferred because quick

change in fan speed is commonly required. This capability will provide for increases in system capacity or pressure requirements due to changes in process, hood design, equipment location, or air cleaning equipment.

5. *Noise*—This is not as important in industrial exhaust situations.
6. *Operating Temperature*—Sleeve bearings are suitable to 250 °F, and ball bearings can be used up to 550 °F. Special cooling devices are required at higher temperatures.
7. *Efficiency*—Select a fan size that will handle the required volume and pressure with minimum horsepower.
8. *Space Limitations.*

6 Ducts

Air conveyed by a duct will impose two loads on the duct's structure. These loads are air pressure and velocity. A duct is a structural assembly, and its optimum construction depends on the maximum loads imposed on it. Usually, duct strength, deflection, and leakage are more functions of pressure rather than velocity.

Static pressure at specific points in an air distribution system is not necessarily the static pressure rating of the fan. Because total pressure decreases in the direction of flow, a duct construction pressure classification equal to the fan outlet pressure (or to the fan total static pressure rating) cannot economically be imposed on the entire duct system. Figure D-28 shows examples of static pressure identification. The static pressure rating changes are shown by "flags" at each point where the duct static pressure classification changes, with the number on the flag indicating the pressure class of the ductwork on each side of the dividing line.

Rectangular Duct Construction

Rectangular duct construction standards provide options for constructing ducts. These include ducts unreinforced and joined by flat type connections only, those joined by flat type joint connectors backed by a qualified reinforcement, those joined by an upright connector that meets reinforcement requirements alone or in conjunction with an incorporated reinforcement, and, in sizes over 48 in. width, those using tie rods that permit the use of smaller reinforcements. Not all options exist at all sizes and all static pressure classes.

Duct construction tables define relationships between static pressure, width, wall thickness, reinforcement spacing, and reinforcement strength so that ducts have adequate strength and acceptable deflection limits. The greater dimension of a duct determines the duct gauge for all four sides. This applies to both reinforced and unreinforced ducts.

The first step in determining construction requirements is to locate the table with the applicable static pressure. The tables that follow can be found in the Sheet Metal and Air Conditioning Contractors National Association, Inc.'s (SMACNA) HVAC Duct Construction Standards. SMACNA provides a number of tables in both U.S. and metric units for different pressure classes. This discussion will refer to the 1 in. w.g. pressure class table in the ensuing paragraphs.

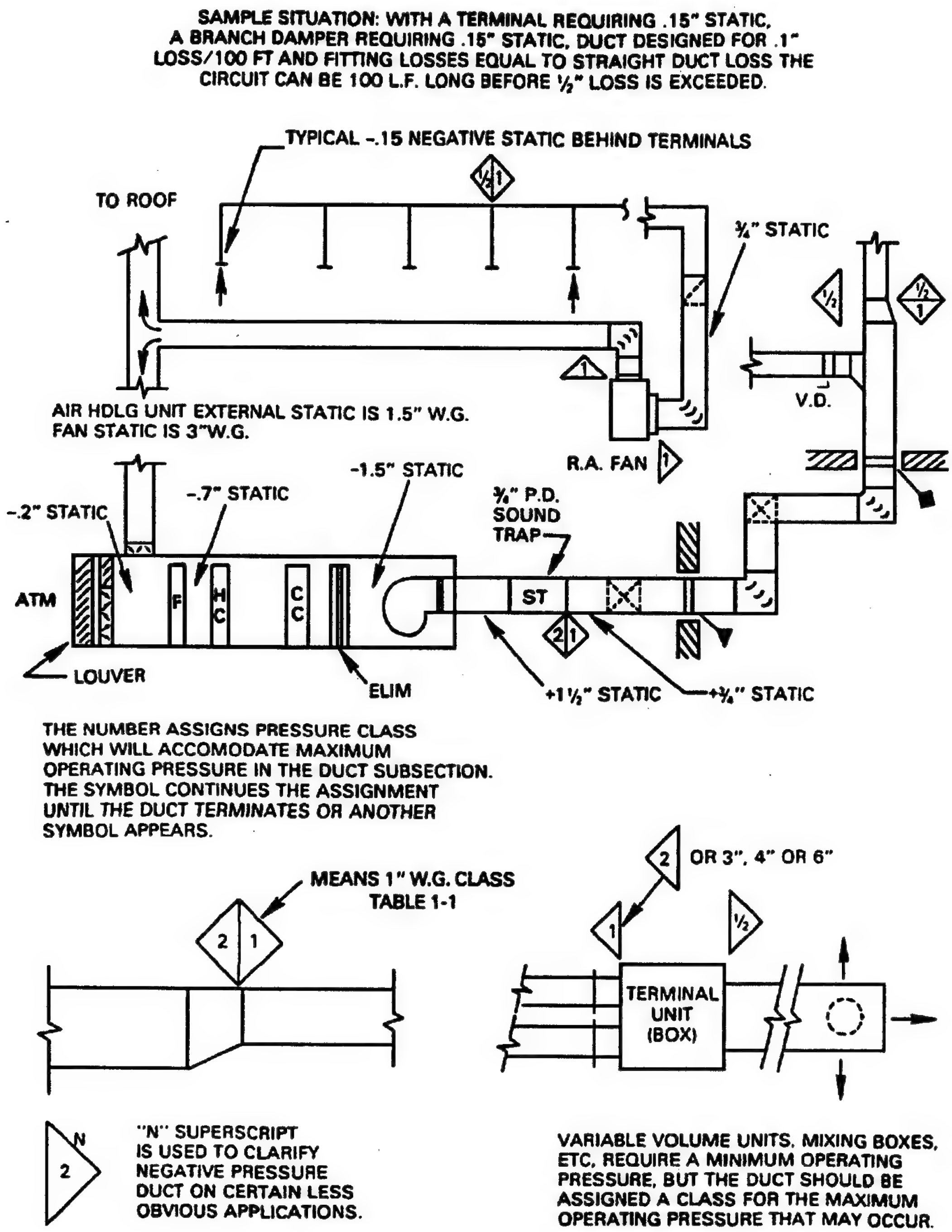


Figure D-28. Duct Pressure Class Designation.

SMACNA - *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

Duct Reinforcing

To find the minimum gauge of the duct and the minimum grade of reinforcement in Table D-10, determine the duct dimension in column 1 and go to the appropriate reinforcement spacing in columns 3 through 10. The minimum grade of reinforcement is given by the alphabet letter, and the minimum gauge of the duct is shown numerically. Each column is an alternative construction available for selection.

The duct side with the greater dimension is investigated first because this side dictates the duct gauge. Then the lesser duct dimension is found in column 1, and the duct gauge used for the wide side is located on the same line. If the duct gauge is in column 2, no reinforcement is required on that side; otherwise, the minimum reinforcement code is the alphabet letter listed under the spacing actually used. The actual duct gauge may occur in a column giving an allowable spacing greater than will be used. In such a case, the minimum reinforcement grade is that associated with the actual spacing.

Transverse Joint and Intermediate Reinforcement

The reinforcement spacing in Table D-10 denotes the distance between two joints or two intermediate reinforcements, or from a joint to an intermediate member. Any joint or reinforcement member having a corresponding letter code like those in Tables D-11 and D-12 may be used as shown in the examples below for various duct sizes (Examples 1 through 3).

The letter coding for reinforcement corresponds to a stiffness index number (EI) that is the modulus of elasticity multiplied by a moment of inertia that is appropriately based on contributing elements of the connector, the reinforcement, the duct wall, or combinations therein.

Example 1 — 18 x 12 in. duct

For a duct fabricated out of 22 ga sheet metal, column 2 of Table D-10 shows that it may be unreinforced.

If the duct is of 24 ga, the 12-in. side is unreinforced, while grade B joints are required at 10 ft minimum spacing on the 18-in. sides. Also, Table D-12 allows the T-1 drive slip to be used on the 18-in. sides. Any joint used on the 18-in. side must meet grade B regardless of joint spacing.

1" W.G. STATIC POS.OR NEG.	TABLE 1-4 RECTANGULAR DUCT REINFORCEMENT								
	NO REINFORCE-MENT REQUIRED	REINFORCEMENT CODE FOR DUCT GAGE NO.							
		REINFORCEMENT SPACING OPTIONS							
DUCT DIMENSION		10'	8'	6'	5'	4'	3'	21/2'	2'
1	2	3	4	5	6	7	8	9	10
10"dn	26 ga.	NOT REQUIRED							
11, 12"	26 ga.								
13, 14"	24 ga.	B-26	B-26	B-26	B-26	B-26	A-26	A-26	A-26
15, 16"	22 ga.	B-24	B-26	B-26	B-26	B-26	B-26	B-26	A-26
17, 18"	22 ga.	B-24	B-26	B-26	B-26	B-26	B-26	B-26	B-26
19, 20"	20 ga.	C-24	C-26	C-26	C-26	C-26	B-26	B-26	B-26
21, 22"	18 ga.	C-24	C-24	C-26	C-26	C-26	B-26	B-26	B-26
23, 24"	18 ga.	C-24	C-24	C-26	C-26	C-26	C-26	B-26	B-26
25, 26"	18 ga.	D-22	D-24	C-26	C-26	C-26	C-26	C-26	B-26
27, 28"	16 ga.	D-22	D-24	D-26	C-26	C-26	C-26	C-26	C-26
29, 30"	16 ga.	E-22	D-24	D-26	D-26	C-26	C-26	C-26	C-26
31-36"	NOT DESIGNED	E-20	E-22	E-24	D-24	D-26	C-26	C-26	C-26
37-42"		F-18	F-20	E-22	E-24	E-26	D-26	D-26	C-26
43-48"		G-16	G-18	F-20	F-22	E-24	E-26	E-26	D-26
49-54"		H-16	H-18	G-20	F-22	F-24	E-24	E-24	E-24
55-60"			H-18	G-20	G-22	F-24	F-24	E-24	E-24
61-72"				H-18G	H-18G	H-22G	F-24	F-24	F-24
73-84"				I-16G	I-18G	I-20G	H-22G	H-22G	G-22
85-96"					I-16H	I-18H	I-20G	H-20G	H-22G
97-108"						I-18G	I-18G	I-18G	I-18G
109-120"							I-18H	I-18H	I-18G

See page 1-15. Circles in the Table denotes only column numbers. For column 2, *see* Fig. 1-7. For columns 3 through 9, *see* Introduction to Schedules. The number in the box is minimum duct gage; the alphabet letter is the minimum reinforcement grade for joints and intermediates occurring at a maximum spacing interval in the column heading. A letter to the right of the gage gives a tie rodded reinforcement alternative. A "t" compels use of tie rod(s) for the reinforcement listing. For beading or crossbreaking, *see* Fig. 1-8.

Table D-10. Rectangular Duct Reinforcement.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

TABLE 1-24 UNREINFORCED DUCT (WALL THICKNESS)							
DUCT DIMENSION	PRESSURE CLASS (In W.G.) Positive or Negative						
	1/2"	1"	2"	3"	4"	6"	10"
8" dn	26	26	26	24	24	24	22
9, 10"	26	26	26	24	22	20	18
11, 12"	26	26	24	22	20	18	16
13, 14"	26	24	22	20	18	18	
15, 16"	26	22	20	18	18	16	
17. 18"	26	22	20	18	16		
19, 20"	24	20	18	16			
21, 22"	22	18	16	16			
23, 24"	22	18	16	16			
25, 26"	20	18					
27, 28"	18	16			Reinforcement is required.		
29, 30"	18	16					
31-36"	16						

This table gives minimum duct wall thickness (gage) for use of flat type joint systems. Plain S and hemmed S connectors are limited to 2" w.g. maximum. Slips and drives must not be less than two gages lighter then the duct wall nor below 24 gage. Double S slips must be 24 gage for ducts 30" wide or less and 22 gage for greater width.

Duct Gage	26 to 22	20	18	16
Minimum Flat Slip and Drive Gage	24	22	20	18

See Figure 1-7 for joint types.

Table D-11. Unreinforced Duct.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

If the duct is of 26 gauge, the 12-in. side is unreinforced, but the 18-in. side has a maximum reinforcement spacing of 8 ft and the minimum size is grade B. Table D-12 allows the T-1 drive slip to be acceptable (up to 20 in. width and 8 ft spacing).

Example 2 — 30 x 18 in. duct

The choices for the 30-in. side are: 16 ga for unreinforced; grade E on 22 ga at 10 ft or D on 24 ga at 8 ft. For the 18-in. side, the choices are the same as outlined in Example 1 for 18-in. width.

TABLE 1-25 T-1 FLAT DRIVE ACCEPTED AS REINFORCEMENT								
DUCT WALL	**26 ga**		**24 ga**		**22 ga**		**20 ga or Heavier**	
Static Pressure	**Maximum Duct Width (W) and Maximum Reinforcement Spacing (S)**							
	W	S	W	S	W	S	W	S
1/2" w.g.	20" 18"	10' N.R.	20"	N.R.	20"	N.R.	20"	N.R.
1" w.g.	20" 14" 12"	8' 10' N.R.	20" 14"	8' N.R.	20" 18"	10' N.R.	20"	N.R.
2" w.g.	18"	5'	18" 12"	8' N.R.	18" 14"	10' N.R.	18"	N.R.
3" w.g.	12" 10"	5' 6'	18" 10"	5' N.R.	18" 12"	5' N.R.	18" 14"	6' N.R.
4" w.g.	Not Accepted		16" 8"	5' N.R.	12" 8"	6' N.R.	12"	N.R.
6" w.g.			12" 8"	5' N.R.	12" 8"	5' N.R.	12" 10"	6' N.R.
Although the flat drive slip T-1 does not satisfy the EI calculation requirements for Classes A, B or C reinforcement, tests predict its suitability for use as reinforcement within the limits of the table.								

N.R. --No reinforcement is required; however, the T-1 Joint may be used.

Table D-12. T-1 Flat Drive Accepted as Reinforcement.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

Example 3 — 54 x 30 in. duct, 5-ft joint spacing preselected

For 54-in. width, grade F at 5 ft is required if 22 ga is selected. A 24-ga duct may be used, but with 4-ft joint spacing).

For the 30-in. side, grade E is required (for 10 ft maximum spacing) on any duct gauge less than 16.

Duct Materials

A variety of materials can be used in the construction of ducts. Selection of the materials used throughout the duct system should be given careful consideration. Consideration must also be given to selection of duct components other than those materials used for the duct walls. Such items include duct liners, pressure sensitive tapes, sealants, adhesives, reinforcements, hangers, etc.

Materials and the construction of exhaust ductwork and fans depends on:

1. Nature of the hood effluents
2. Surrounding temperature
3. Lengths and arrangement of duct runs
4. Flame and smoke spread rating
5. Duct velocities and pressures.

Both present and future effluents should be evaluated when selecting duct materials and construction. These effluents may be classified as organic or inorganic chemical gases, vapors, fumes, or smokes. Exhaust system fans, ducts, and coatings can be damaged by these effluents through corrosion, dissolution, and melting.

The condensation of vapors in the exhaust system is affected by the surrounding temperature of the space in which the ductwork and fans are located. Condensation contributes to the corrosion of metals with or without the presence of chemicals.

When duct runs are short and direct, and when the air is maintained at reasonable (higher) velocities, the chance of attack by effluents is less. The longer the duct, the longer will be the period of exposure to effluents and the greater the degree of condensation. Horizontal runs provide surfaces where moisture can remain longer than it may on vertical runs. If condensation is probable, sloped ductwork and condensate drains should be provided.

Fan operation may be continuous or intermittent. Intermittent fan operation allows longer periods of wetness because of condensation.

Following is a list of duct materials and their characteristics:

- *Galvanized Steel* - Widely used as a duct material for most air handling systems; not recommended for corrosive product handling or temperatures above 400 °F. Advantages include high strength, rigidity, durability, rust resistance, availability, nonporosity, workability, and weldability. Galvanized sheets with the surface treated for painting are commonly used.

- *Carbon Steel (Black Iron)* - Applications include flues, stacks, hoods, other high temperature duct systems, kitchen exhaust systems, and ducts requiring paint or special coating. Advantages include high strength, rigidity,

durability, availability, paintability, weldability, and nonporosity. Some limiting characteristics are corrosion resistance and weight.

- *Aluminum* - Aluminum can be used in duct systems for moisture laden air, louvers, special exhaust systems, ornamental duct systems, and is often substituted for galvanized steel in HVAC duct systems. Some advantages include weight, resistance to moisture corrosion, and availability. Limiting characteristics include low strength, material cost, weldability, and thermal expansion.

- *Stainless Steel* - Used in duct systems for kitchen exhaust, moisture laden air, and fume exhaust. Advantages include high resistance to corrosion from moisture and most chemicals and the ability to take a high polish. Limiting characteristics include labor and material costs, workability, and availability.

- *Copper* - Copper applications include duct systems exposed to outside elements and moisture laden air, certain chemical exhaust, and ornamental ductwork. Advantages are durability and corrosion resistance and that it accepts solder readily and is nonmagnetic. Limiting characteristics are cost, ductility, electrolysis, thermal expansion, and stains.

- *Fiberglass Reinforced Plastic (FRP)* - Applications include chemical exhaust, scrubbers, and underground duct systems. Resistance to corrosion and ease of modification are advantages of FRP. Limiting characteristics include cost, weight, range of chemical and physical properties, brittleness, fabrication (necessity of molds and expertise in mixing basic materials), and code acceptance.

- *Polyvinyl Chloride (PVC)* - Applications are exhaust systems for chemical fumes and hospitals, and underground duct systems. Advantages include resistance to corrosion, weight, weldability, and ease of modification. Limiting characteristics include cost, fabrication, code acceptance, thermal shock, and weight.

- *Polyvinyl Steel (PVS)* - Applications include underground duct systems, moisture laden air, and corrosive air systems. Some advantages are resistance to corrosion, weight, workability, fabrication, and rigidity. Some limiting characteristics include temperature limitations (250 °F maximum), weldability, code acceptance, and susceptibility to coating damage.

- *Concrete* - Concrete can be used for underground ducts and air shafts. Advantages include compressive strength and corrosion resistance. Cost, weight, porosity, and fabrication (requires forming processes) are some limiting characteristics.

- *Rigid Fibrous Glass* - Most widely used in interior HVAC low pressure duct systems. Advantages include weight, thermal insulation and vapor barrier, acoustical qualities, ease of modification, and inexpensive tooling for fabrication. Limiting characteristics include cost, susceptibility to damage, system pressure, and code acceptance.

- *Sheetrock* - Applications include ceiling plenums, corridor ducts, and air shafts. Cost and availability are advantages, while weight, code acceptance, and leakage are limiting characteristics.

7 Air Cleaners

Air cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet variations in air cleaning requirements. The degree of removal required, quantity and characteristics of the contaminant to be removed, and conditions of the air or gas stream will all have a bearing on the device selected for any given application. Air cleaning equipment is usually selected to:

1. Conform to Federal, state, or local emission standards and regulations.
2. Prevent reentrainment of contaminants to work areas where they may become a health or safety hazard.
3. Reclaim usable materials.
4. Permit cleaned air to recirculate to work spaces and/or processes.
5. Prevent physically damaging adjacent property.

For particulate contaminants, air cleaning devices are divided into two basic groups: air filters and dust collectors.

Air Filters

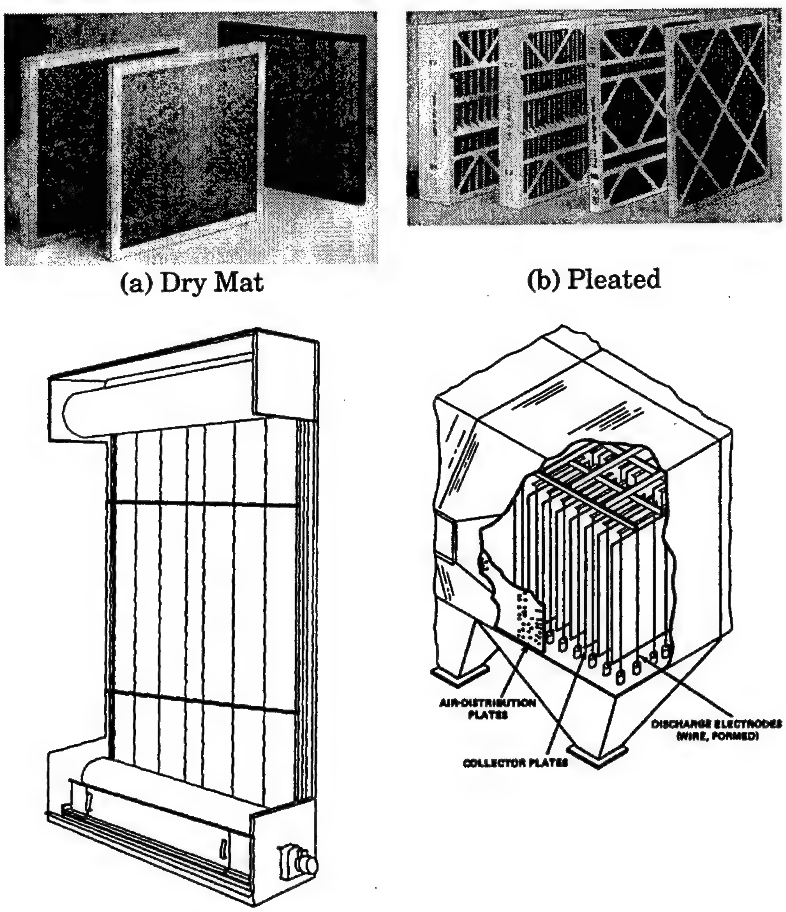
Air filters are designed to remove dust concentrations of the order found in outside air, and are used in ventilation, air conditioning, and heating systems where dust loading seldom exceed one grain per thousand cubic feet of air and is usually well below 0.1 grains per thousand feet of air.* All of the common types of air filters fall into three broad categories: (1) fibrous media, (2) renewable media, and (3) electronic air cleaners.

* A grain is a unit of weight measure and is equivalent to 1/7000 of a pound.

Fibrous media unit filters accumulate dust loads that cause pressure drop to increase up to some maximum permissible value. During this period of increase, efficiency also increases. At high dust loads, however, dust may adhere poorly to the filter's fibers, causing efficiency to drop. Filters in this condition should be replaced or reconditioned. This category includes viscous impingement and dry type air filters.

Another category is the renewable media filter in which fresh media is introduced into the air stream to maintain nearly constant resistance. These filters maintain nearly constant efficiency.

Electronic air cleaners have essentially constant pressure drop and efficiency, unless their precipitating elements become severely dust loaded. Figure D-29 shows four basic types of air filters: dry mat, pleated, roll type, and electrostatic.



(a) Dry Mat (b) Pleated

(c) Roll Type (d) Electrostatic

Figure D-29. Air Filter Types.

Source for a-c: *NAFA Guide to Air Filtration,* 1993, used with permission.
Source for d: 1996 *ASHRAE Systems and Equipment Handbook.*

Manufacturers' ratings should be used for the type of filtering medium selected. Filters are usually selected for ease of maintenance and to provide the highest degree of air cleanliness feasible or required by the installation.

Dust Collectors

Dust collectors are usually designed for the much heavier loads from work shops or industrial processes where the air or gas to be cleaned originates in local systems or process stack gas effluents. Loading will vary from less than 0.1 to 20 grains or more per cubic foot. Therefore, dust collectors are and must be capable of handling concentrations some 100 to 20,000 times greater than air filters.

Dust collection equipment is available in numerous designs using a number of principles and featuring wide variation in effectiveness, first cost, operating and maintenance cost, space, arrangement, materials, and construction. Consultation with the equipment manufacturer is the recommended procedure in selecting a collector for any problem where extensive previous plant experience on the specific dust problem is not available. Factors influencing equipment selection include:

- Concentration and particle size of contaminant
- Degree of collection required
- Characteristics of air or gas stream
- Characteristics of contaminant
- Energy requirements
- Method of dust disposal.

The five basic types of dust collectors available are electrostatic precipitators, fabric filter, unit collector, wet collector, and dry centrifugal collector.

Electro-Static Precipitators

In electrostatic precipitation, a high potential electric field is established between discharge and collecting electrodes of opposite polarity. The discharge electrode is of small cross-sectional area, such as a wire or piece of flat stock, and the collection electrode is large in surface area, such as a plate.

The stream to be cleaned passes through an electrical field that develops between the electrodes. At a critical voltage, the molecules are separated into positive and negative ions. This is called "ionization" and takes place at or near the surface of the discharge electrode. Ions, having the same polarity as the discharge electrode, attach themselves to neutral particles in the gas stream as they flow through the precipitator. They are then attracted to the collecting plate, which is of opposite polarity. Upon contact with the collecting surface, dust particles lose their charge and can be easily removed by vibration, washing, or by gravity (Figure D-30).

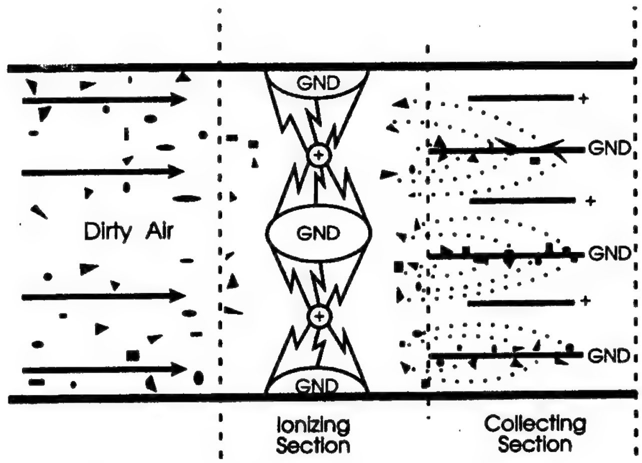


Figure D-30. Electrostatic Precipitator.

Source: *NAFA Guide to Air Filtration,* 1993. Used with permission.

The electrostatic process consists of:

1. Ionization of the gas
2. Charging the dust particles
3. Transportation of the particles to the collecting surface
4. Neutralization, or removing the charge from the dust particle
5. Removal of the particle from the collection surface.

Fabric Filter Collectors

Fabric filter collectors remove particulates from carrier gas streams by interception, impaction, and diffusion mechanisms. The fabric may be constructed of a variety of materials, and may be woven or non-woven. A heavy non-woven fabric is more efficient than a woven fabric since the void areas or pores in the felted fabric are smaller. Fabric collectors are not 100 percent efficient, but well-designed, adequately sized, and properly operated fabric collectors can be expected to operate at efficiencies in excess of 99 percent. Commercially available fabric collectors have fabric configured as tubes or stockings, envelopes, or pleated cartridges.

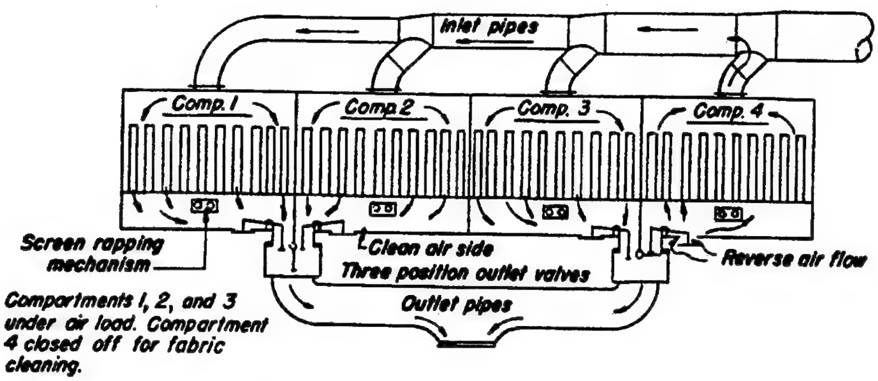


Figure D-31. Multiple-Section, Continuous-Duty, Automatic Fabric Collector.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Unit Collectors

Unit collectors (Figure D-32) have small fabric filters and capacities in the range of 200 to 2000 cfm. They have integral air movers, small space requirements, and simplicity of installation. In most applications, cleaned air is recirculated although discharge ductwork may be used if the added resistance is within the capability of the air mover.

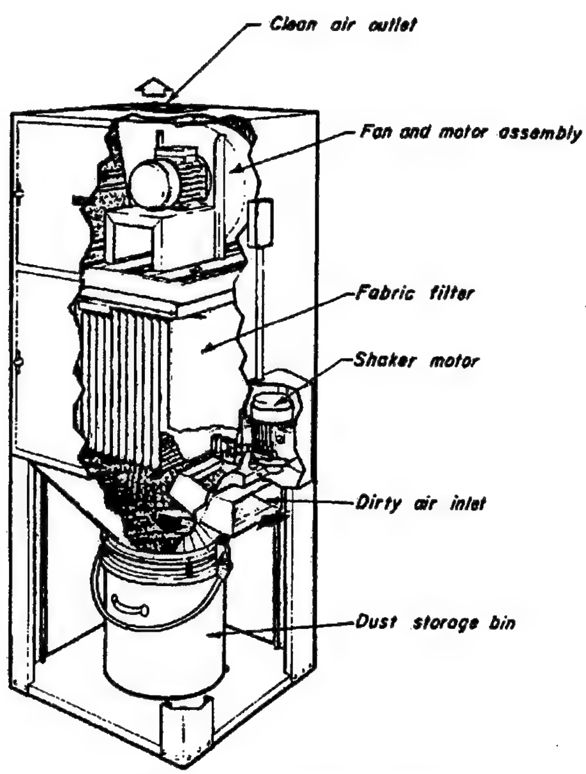


Figure D-32. Unit Collector.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Unit collectors are often used in the metal working industry to fill the need for dust collection from isolated, portable, intermittently used, or frequently relocated dust producing operations. Typically, a single collector serves a single dust producing operation with the energy saving advantage that the collector need operate only when the dust producing machine is in operation.

Wet Collectors

Wet collectors are commercially available in many different designs. These collectors have the ability to handle high temperature and moisture-laden gases. Wet collectors have one characteristic not found in other collectors—their ability to humidify. Humidification, the process of adding water vapor to the air stream through evaporation, may be either advantageous or disadvantageous depending on the situation.

Chamber or spray tower collectors. These consist of a round or rectangular chamber into which water is introduced via spray nozzles. The principal mechanism of these collectors is impaction of dust particles on the liquid droplets created by the nozzles. These droplets are separated from the air stream by centrifugal force or impingement on water eliminators.

Packed tower collectors. These collectors are essentially contact beds through which gases and liquid pass either concurrently, counter-currently, or in crossflow. They are used primarily for applications involving gas, vapor, and mist removal (Figure D-33).

Wet centrifugal collectors. Wet centrifugal collectors comprise a large portion of the commercially available designs. This type uses centrifugal force to accelerate the dust particle and impinge it upon a wetted collector surface (Figure D-34).

Wet dynamic precipitators. These use water sprays within a fan housing, and obtain precipitation of the dust particles on the wetted surfaces of an impeller with a special fan blade shape (Figure D-35).

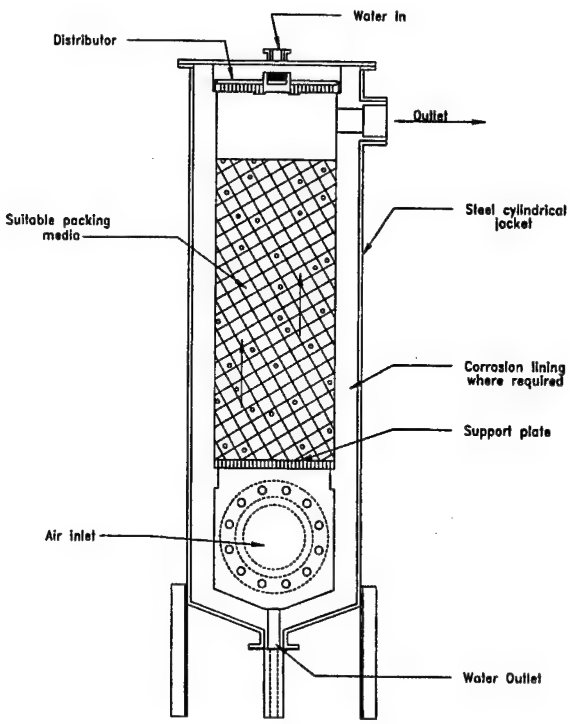
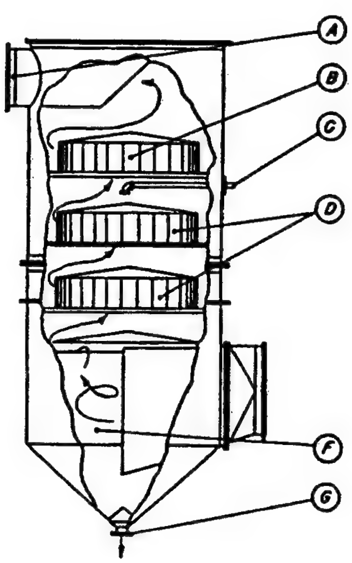


Figure D-33. Packed Tower Collector.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.



Symbols	*Parts*
A	*Clean air outlet.*
B	*Entrainment separator.*
C	*Water inlet.*
D	*Impingement plates.*
E	*Dirty air inlet.*
F	*Wet cyclone for collecting heavy material.*
G	*Water and sludge drain.*

Figure D-34. Wet Centrifugal Collectors.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

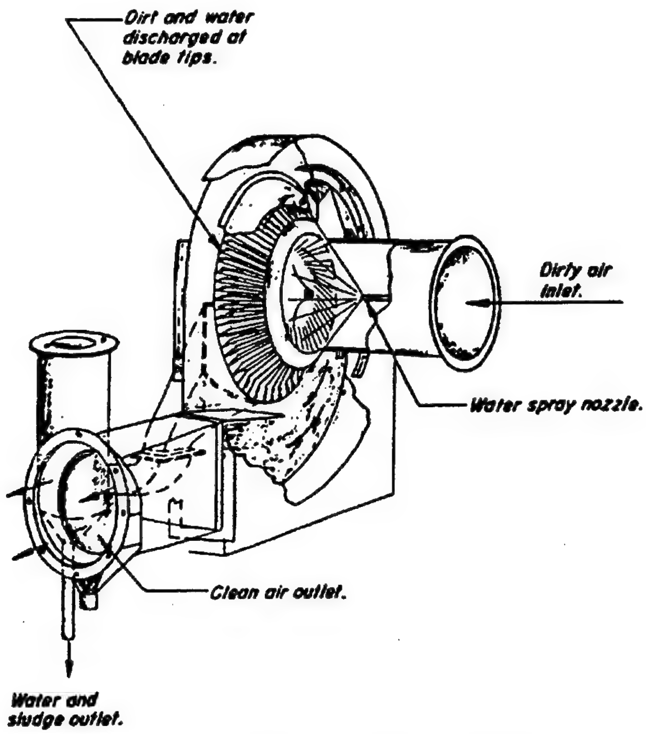


Figure D-35. Wet Dynamic Precipitator.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Orifice type collectors. Orifice type collectors bring the air flow in contact with a sheet of water in a restricted passage. Water flow may be induced by the velocity of the air stream or maintained by pumps and weirs.

Venturi collectors. Venturi collectors use a venturi-shaped constriction to establish throat velocities considerably higher than those experienced with the orifice type. The collection mechanism of the venturi is impaction.

Dry Centrifugal Collectors

Dry centrifugal collectors can be divided into two basic groups categorized by their effectiveness in removal of smaller dust particles.

Cyclone collectors. The cyclone collector is commonly applied for the removal of coarse dusts from an air stream, as a pre-cleaner to more efficient dry or wet dust collectors, and/or as a separator in product conveying systems using an air stream to transport material. Its principal advantages are low cost, low maintenance, and low pressure drop, but it cannot be used for collection of fine particles.

High efficiency centrifugal collectors. In these collectors, higher centrifugal forces are exerted on dust particles in a gas stream. Centrifugal force is a function of peripheral velocities and angular acceleration. Improvement in dust separation efficiency has been obtained by increasing velocities through a cyclone shaped collector using a skimmer or other design feature, with a number of small diameter cyclones in parallel, and placing units in series in some unusual applications (Figure D-36).

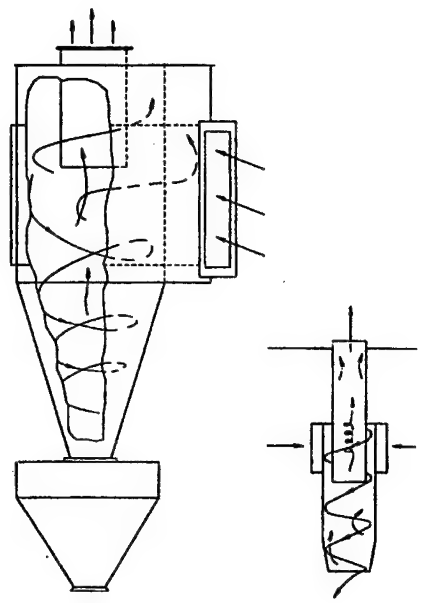


Figure D-36. High Efficiency Centrifugal Collector.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

8 Exhaust Hoods

A hood can be defined as "a ventilated enclosed work space intended to capture, contain, and exhaust fumes, vapors, and particulate matter generated inside the enclosure. It consists basically of side, back, and top enclosure panels, a work surface or counter top, an access opening called the face, a sash, and an exhaust plenum equipped with a baffle system for the regulation of air flow distribution." Listed below are some definitions that should help in the understanding of exhaust hoods (Figure D-37):

- Capture Velocity—The air velocity at any point in front of the hood or at the hood opening necessary to overcome opposing air currents, and to capture the contaminated air at that point by causing it to flow into the hood.
- Face Velocity—Air velocity at the hood opening.
- Slot Velocity—Air velocity (fpm) through the openings in a slot type hood. It is used primarily as a means of obtaining uniform air distribution across the face of the hood.
- Plenum Velocity—Air velocity (fpm) in the plenum. For good air distribution with slot type hoods, the maximum plenum velocity should be half of the slot velocity or less.
- Duct Velocity—Air velocity (fpm) through the duct cross-section. When solid material is present in the air stream, the duct velocity must be equal to the minimum design duct velocity.
- Minimum Design Duct Velocity—Minimum air velocity (fpm) required to move the particulates in the air stream.

Hoods can be classified as either enclosed or nonenclosed (Figure D-38). Enclosed hoods provide a more economical contaminant control because the effects of room air currents and the exhaust rate are small compared to those for a nonenclosed hood. When nonenclosed hoods need to be used, careful attention

should be given to air flow patterns around the process and hood. Nonenclosed hoods should also be located so that the contaminant is drawn away from an operator's breathing zone. Figure D-39 illustrates the various hood types available.

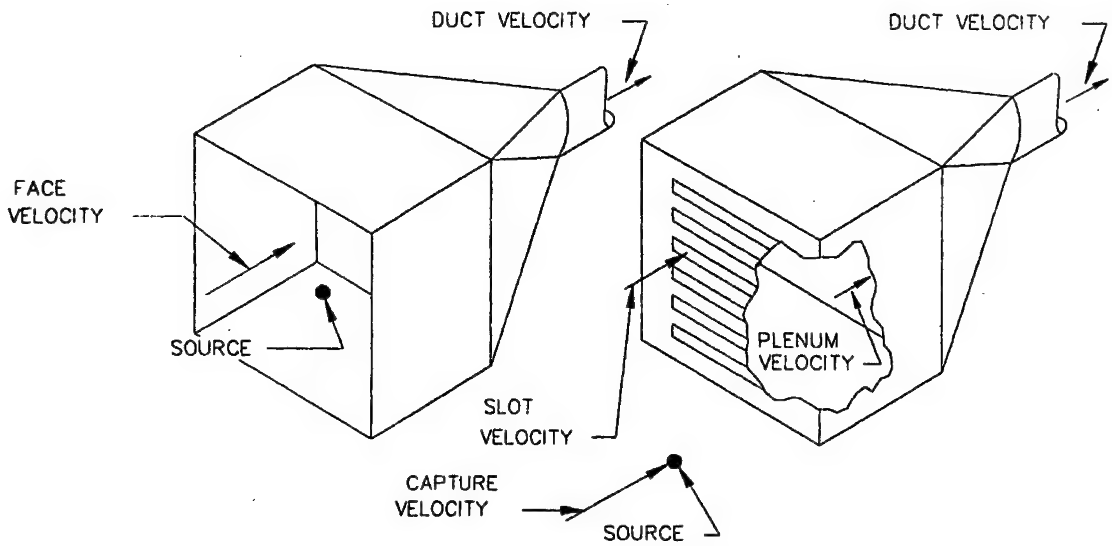


Figure D-37. Basic Exhaust Hood Terms.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice*, 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

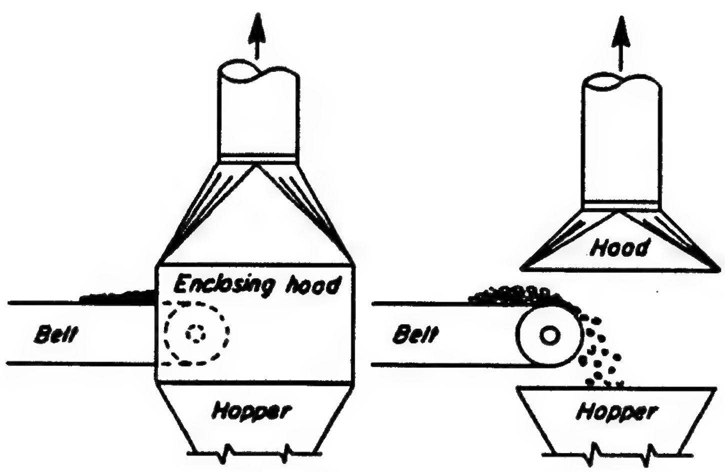


Figure D-38. Enclosed Hood (left) and Nonenclosed Hood (right).

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice*, 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Hood access openings should be as small as possible. Access should be provided for inspection and maintenance. Hoods should be placed as close as possible to the source of the contaminant. The required volume varies with the square of he distance from the source.

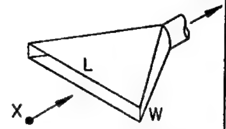
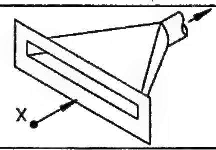
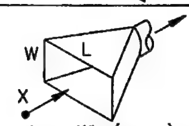
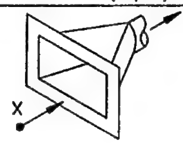
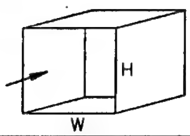
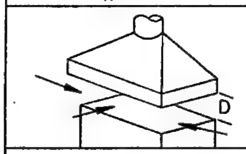
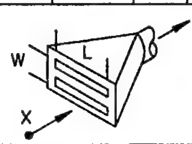
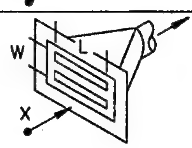
HOOD TYPE	DESCRIPTION	ASPECT RATIO, W/L	AIR FLOW
	SLOT	0.2 OR LESS	Q = 3.7 LVX
	FLANGED SLOT	0.2 OR LESS	Q = 2.6 LVX
 A = WL (sq.m)	PLAIN OPENING	0.2 OR GREATER AND ROUND	$Q = V(10X^2 + A)$
	FLANGED OPENING	0.2 OR GREATER AND ROUND	$Q = 0.75V(10X^2 + A)$
	BOOTH	TO SUIT WORK	Q = VA = VWH
	CANOPY	TO SUIT WORK	Q = 1.4 PVD SEE VS-99-03 P = PERIMETER D = HEIGHT ABOVE WORK
	PLAIN MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = V(10X^2 + A)$
	FLANGED MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = 0.75V(10X^2 + A)$

Figure D-39. Hood Types.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Capture Velocity

Table D-13 shows ranges and applications of capture velocities for several industrial operations. These capture velocities are based on successful experience under ideal conditions.

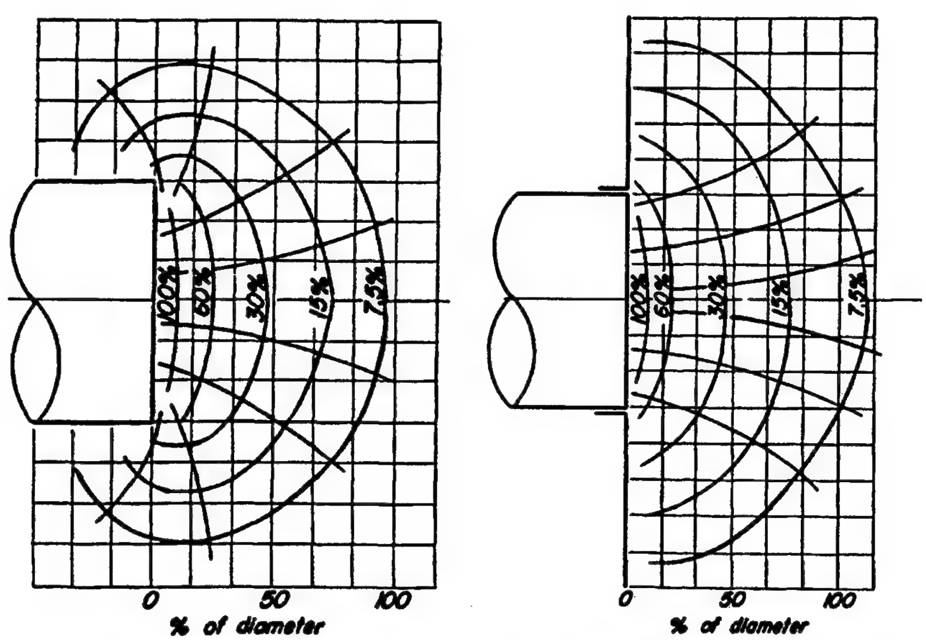
Condition of Dispersion of Contamination	Example	Capture Velocity (fpm)
Released with practically no velocity into quiet air.	Evaporation from tanks; degreasing, etc.	50-100
Released at low velocity into moderately still air.	Spray booths; intermittent container filling; low speed conveyor transfers; welding; plating; pickling	100-200
Active generation into zone of rapid air motion.	Spray painting in shallow booths; barrel filling; conveyor loading; crushers	200-500
Released at high initial velocity into zone at very rapid air motion.	Grinding; abrasive blasting; tumbling	500-2000
In each category above, a range of capture velocity is shown. The proper choice of values depends on several factors:		
Lower End of Range	*Upper End of Range*	
1. Room air currents minimal or favorable to capture.	1. Disturbing room air currents.	
2. Contaminants of low toxicity or of nuisance value only.	2. Contaminants of high toxicity.	
3. Intermittent, low production.	3. High production, heavy use.	
4. Large hood-large air mass in motion.	4. Small hood-local control only.	

Table D-13. Capture Velocities.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Flanging

Wherever possible, flanges (which are projected rims or collars) should be provided to eliminate air flow from ineffective zones where no contaminant exists. The hood effectiveness is increased, and air requirements can be reduced by as



(a) Circular Opening (b) Circular Opening with Flange

Figure D-40. Effect of Flanging.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

much as 25 percent. Figure D-40 shows that the lines in front of the hood are lines of equal velocity and are called flow contours. The lines perpendicular to the flow contours are called streamlines. The tangent to a streamline at any point indicates the direction of airflow at that point.

Volumetric Flow Rate

The exhaust volumetric flow rate can be calculated after the capture velocity and hood configuration are determined.

For enclosed hoods, the exhaust volumetric flow rate can be calculated by the equation:

$$Q = V \times A \qquad \text{[Eq D-8]}$$

where: Q = volumetric flow rate, cfm
V = average flow velocity, fpm
A = flow cross-sectional area, sq ft

The inflow velocity is usually around 100 fpm.

For nonenclosed hoods, the capture velocity and the air velocity at the point of contaminant release must be equal and be directed so that the contaminant enters the hood. This results in different volumetric flow rate equations for different types of hoods. For unflanged round and rectangular openings, the required flow rate equation is:

$$Q = V \times [(10X \times 10X) + A]$$

where Q = flow rate, cfm
V = capture velocity, fpm
X = centerline distance from the hood face to the point of contaminant generation, ft
A = hood face area, sq ft
L = long dimension of the slot, ft

For slot hoods, the required flow rate is predicted by an equation for openings between 0.5 to 2 in. in width:

$$Q = 3.7 \times L \times V \times X$$

If a flange is installed around the hood opening, the required flow rate for plain openings is reduced to 75 percent of that for the corresponding unflanged opening. The flange size should be approximately equal to four times the area divided by the perimeter of the face hood. For flanged slots with aspect ratios less than 0.15 and flanges greater than three times the slot width, the equation is:

$$Q = 2.6 \times L \times V \times X$$

A baffle is a solid barrier that prevents airflow from unwanted areas in front of the hood. For hoods that include baffles, the DallaValle half-hood equation is used to approximate the required flow rate:

$$Q = V \times [(5X \times 5X) + A]$$

Volumetric Flow Rate Example

Design a nonenclosed hood to capture a contaminant that is released with a low velocity 2 ft in front of the face of the hood. The hood face dimensions are 1.5 × 4.0 ft. The hood rests on a bench, and a flange is placed on the sides and top of the face. The room air currents are variable in direction but less than 50 fpm, and the contaminant has low toxicity. Determine the volumetric flow rate required to capture the contaminant if the hood is used continuously.

Solution

Table D-13 shows that a capture velocity of 50 to 100 fpm is required. The capture velocity selected must be greater than the room air currents, so 80 fpm will be selected. Modifying the equation listed for baffles so reduction is included for flanges results in:

$$Q = 0.75V \times [(5X \times 5X) + A]$$

$$Q = (0.75 \times 80) \times [(5 \times 2) \times (5 \times 2) + (1.5 \times 4)] = 1{,}560 \text{ cfm}$$

Special Situations

Some operations may require exhaust flow rates different from those listed previously. Some of these reasons are:

1. The exhaust from a hot process requires special consideration because of the heated air effect near the process. Determining the flow rate for this process

requires knowing the conventional heat transfer rate and physical size of the process.

2. High-speed rotating machines such as escaping compressed air from pneumatic tools, high-speed belt material transfer systems, and falling granular materials all produce air currents. The direction and size of the airflow should be taken into consideration when designing the hood.
3. Room air currents caused by spot cooling, compensating air, or cross drafts.
4. Exhaust flow rates that are insufficient to dilute combustible vapor-air mixtures to less than about 25 percent of the lower explosive limit of the vapor.

9 Controls

Automatic control is used to modulate equipment capacity to meet load requirements, and to provide safe operation of equipment. It requires mechanical, electrical, and electronic control devices, and implies that human intervention is limited to starting and stopping equipment, and adjusting control set points.

Components of Automatic Control Systems

Control devices for HVAC systems can be grouped by their function within a complete control system. These groups are:

1. Sensing elements
2. Controllers
3. Controlled devices
4. Auxiliary devices.

Sensing Elements

A sensor is the component in the control system that measures the value of the controlled variable. The controlled variable is the variable such as temperature, humidity, or pressure that is being controlled. The change in the controlled variable produces a change in some physical or electrical property of the primary sensing element, which is then available for translation or amplification by mechanical or electrical signal. Sensors are most often used for temperature, humidity, pressure, and water or fluid flow. Sensors can also be used for flame detection, measurement of smoke density, current, CO_2, or CO.

Controllers

Controllers take the sensor effect, compare it with the desired control condition (set point), and regulate an output signal to cause action on the controlled device. The controller and sensor can be combined in a single instrument such as a room

thermostat, or they can be two separate devices. There are five basic types of controllers:

1. Electric/electronic controllers
2. Indicating or recording controllers
3. Pneumatic receiver controllers
4. Direct digital controllers
5. Thermostats.

Controlled Devices

The controlled device is most frequently used to regulate or vary the flow of air within an HVAC system. Air flow control devices are called dampers. Dampers must be properly sized and selected for a particular application for the control system to control the controlled variable properly. The control system's link to the damper is called an operator or actuator. This device uses electricity, compressed air, or hydraulic fluid as a power source.

Auxiliary Devices

Some examples of auxiliary devices are: transformers, electric relays, potentiometers, manual switches, and auxiliary switches.

Outside Air Control

The total control of an air handling system can be subdivided into several elements including outside air, heating, cooling, humidity, pressure, and space conditions. The control of outside air can be accomplished in many ways.

Fixed Outside Air

This type of control system (Figure D-41) is sized to make up exhaust and exfiltration from the space. Control consists of a two-position outside air damper interlocked to open when the supply fan runs. A manual return air damper usually provides balancing. This system is not energy efficient because it does not use “free” cooling (see ***Economy Cycle***). Most individual room units use this method.

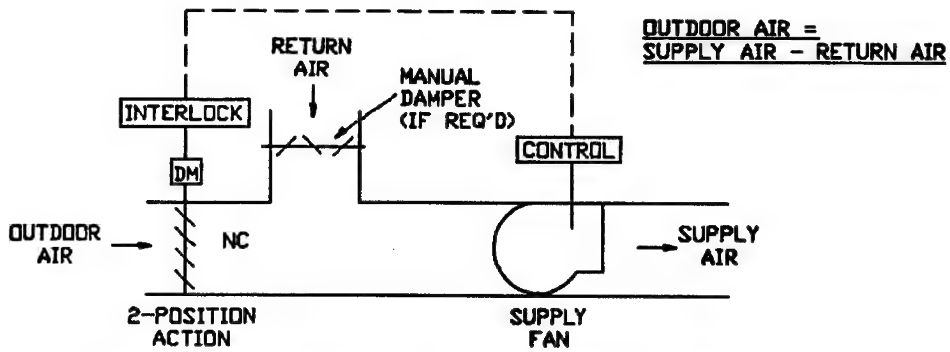


Figure D-41. Fixed Minimum Outdoor Air for Systems without Return Fans.
Reprinted with permission from the 1987 *ASHRAE HVAC Systems and Applications Handbook.*

100% Outside Air

This type of control (Figure D-42) can be used in buildings with large exhaust air requirements. Control consists of a two-position outside air damper interlocked to run when the supply fan runs. Interlocks are also provided between supply and exhaust fans.

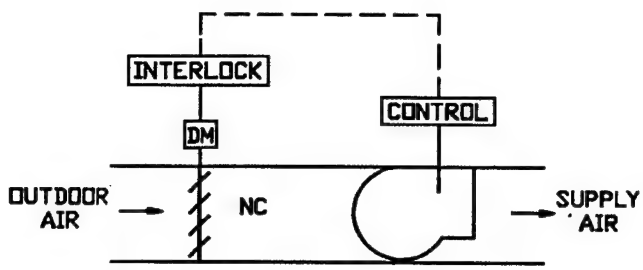


Figure D-42. 100% Outside Air Control.
Reprinted with permission from the 1987 *ASHRAE HVAC Systems and Applications Handbook.*

Economy Cycle

This type of control (Figure D-43) is used most often. A properly designed economy cycle is very energy efficient. When the supply fan is started, the outside air damper opens to the minimum position required for ventilation or exhaust makeup. When the outside air temperature is above a high limit, usually 70 to 75 °F, the outside air damper stays in the minimum position. When the outside air temperature is below the high limit, the outside return and relief air dampers are modulated to maintain a mixed air temperature not less than the low limit set point, usually 55 to 60 °F. In practice, a wide variety of control configurations achieve this sequence.

Thermal energy can be saved by adding reset of the low limit set point in response to cooling load. The set point is reset upward as the cooling load decreases, minimizing reheat energy usage. Many modern systems use this practice, and it is often fairly simple to retrofit older systems.

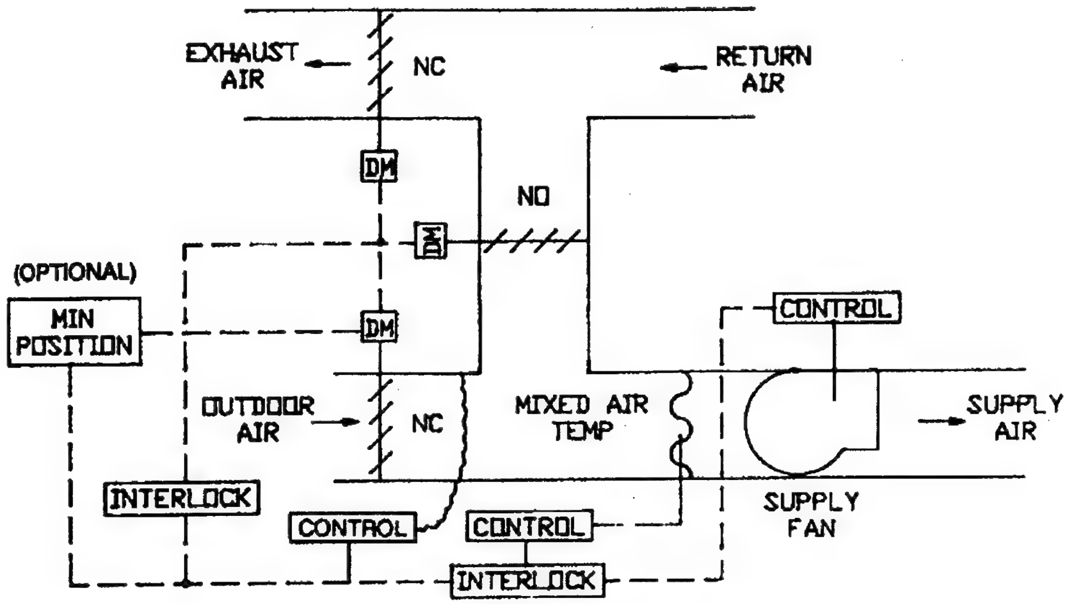


Figure D-43. Economizer Cycle Control.

Reprinted with permission from the 1987 *ASHRAE HVAC Systems and Applications Handbook*.

10 Acceptance Testing

Before starting an acceptance testing procedure, inspections should be made to confirm that equipment has been completely installed, proper electrical connections have been made, automatic controls are complete and operating, and the building is completely closed in with windows, doors, etc.

Prechecks

The following is a preliminary check-out procedure that should be used to confirm that equipment is ready to be tested, adjusted, and balanced:

1. Obtain all equipment data from the manufacturers, and from the design specifications.
2. Obtain and calibrate the instrumentation that will be used.
3. Make sure all measuring points are accessible.
4. Confirm the following on fans:
   a. All bearings have been lubricated.
   b. Fan wheels clear the housing.
   c. All foreign objects have been removed.
   d. Motors have been fastened securely.
   e. All drives have been correctly aligned.
   f. Belt tensions are correct.
   g. Fan rotations are correct.
   h. Duct flexible connections are properly aligned.
   i. Vibration isolators or bases have the correct springs and are in the right location, and that the springs are not collapsed.
   j. Fan flow rate.
5. Confirm the following on all duct systems:
   a. All outside air intake, return air, and exhaust air dampers are in the proper position and operable.
   b. All system volume dampers and fire dampers have been installed, are in the full open position, and are accessible.

c. Inspect access doors and hardware for tightness and leakage, and verify that all necessary access doors have been installed.
d. Openings have been provided in walls and plenums for proper air passage.
e. Duct flow rates.
f. Backdraft dampers installed and operational.
g. Determine duct leakage.

6. Locate all start-stop, disconnect switches, electrical interlocks, and motor starters.

The following is a checklist to follow during acceptance testing.

EXHAUST SYSTEM ACCEPTANCE TESTING CHECKLIST

PROJECT:____

LOCATION:____

NAME:____

A. Prechecks	**Checked**	**Date Checked**
1. All equipment data received from the manufacturer and design specifications		
2. All instrumentation obtained and calibrated		
3. All measuring points accessible		
4. All start-stop, disconnect switches, electrical interlocks, and motor starters located		

B. Fans		**Correct** yes	no	**Date Checked**
1. All bearings lubricated				
2. Fan wheels clear housing				
3. All foreign objects removed				
4. Motors fastened securely				
5. All drives correctly aligned				
6. Belt tensions				
7. Fan rotations				
8. Duct flexible connections properly aligned				
9. Vibration isolators or bases have correct springs, are in right locations, and are not collapsed				
	Design	TAB	Actual	
10. Correct flow rate (cfm)				

C. Ducts		**Correct** yes	no	**Date Checked**
1. All outside air intake, return air, and exhaust air dampers are in proper position and operable				
2. All system volume dampers and fire dampers installed, in full open position and operable				
3. All necessary access doors are installed, tight, and free of leakage				
4. Openings provided in walls and plenums for proper air passage				
	Design	TAB	Actual	
5. Selected duct flow rate (cfm)				

After confirming that the preliminary check-out procedures have been completed, the following procedures can be reviewed. The purpose of these checks is to provide the acceptance testing team with a concise outline of what the TAB contractor was supposed to have done during TAB.

Basic to All Air Systems

1. Confirm that every item affecting the air flow of a duct system is ready. (Windows and doors are closed, ceiling tiles are in place, etc.).
2. Confirm that all automatic control devices did not affect TAB operations.
3. Establish the conditions for the maximum demand system air flow.
4. After verifying that all dampers are open or set, start all related systems (return, exhaust, etc.) and the system being balanced with each fan running at the design speed. Upon starting each fan, immediately check the fan motor amperage. If the amperage exceeds the nameplate full load amperage, stop the fan to determine the cause or make the necessary adjustments.
5. Confirm that all related system fans serving each area within the space being balanced are operating.
6. If a supply fan is connected to a return air system and an outside air intake, set all system dampers and controls so that the air returned from the individual rooms or areas supplied by the fan is returned via the related return air system. Normally this will involve opening an outside air damper to the minimum position, opening the return air damper, and closing exhaust air and relief air dampers.
7. Determine the volume of air being moved by the supply fan at design rpm by one or more of the following methods:

a. Perform a pitot tube traverse of the main duct or ducts leaving fan discharge.

b. Verify fan curves or fan performance charts. To do this, amperage, voltage, and static pressure readings need to be taken.

8. If the supply fan volume is not within ±10 percent of the design capacity at design rpm, determine the reason by reviewing all system conditions, procedures, and recorded data. Check and record the air pressure drop across filters, coils, eliminators, sound traps, etc. to see if excessive loss is occurring. Study duct and casing conditions at the fan inlet and outlet.
9. Using the methods outlined in paragraph 8, determine the volume of air being handled by a return air fan, if used; if a central exhaust fan system is used, also determine the cfm being handled by the exhaust fan. If several exhaust fans are related to a supply system, it is not generally necessary to

measure the cfm of each exhaust fan until after the supply system is balanced.

10. If the measured cfm of the supply fan, central return fan, or central exhaust fan varies more than 10 percent from design, adjust the drive of each fan to obtain the approximate required cfm. Confirm that the fan motor is not overloaded.
11. Make a preliminary survey by spot checking air circulation in various rooms. With knowledge of the supply and return or exhaust fan volumes and data from the survey, determine if the return or exhaust air system should be balanced before the supply system is balanced. The assumption is made that the supply system is not restrained by the exhaust system or the return system. However, if such a restraint exists, the exhaust system or the return system should be balanced prior to continuing with the supply system.
12. The system is considered balanced in accordance with these procedural standards when the value of the air quantity of each inlet or outlet device is measured and found to be within 10 percent of the design air quantities.

Exhaust and Return Air Systems

1. Follow procedures 1 through 7 under the previous section.
2. Determine the volume of air being moved by the exhaust fan at design rpm by fan curves, or by pitot tube traverse of the main duct or the ducts leaving the fan discharge.
3. The exhaust fan volume should be within ±10 percent of the design capacity if earlier procedures were followed. Check and record the air pressure drop across filters, coils, sound traps, etc., to see if any excessive loss is occurring. Study duct and casing conditions at the fan inlet and outlet. Record the exhaust fan suction static pressure, fan discharge static pressure, amperage, and cfm measurements. Confirm that the fan motor is not overloaded.
4. If the exhaust system is being balanced before the supply and/or return air systems, and if the measured cfm of any fan varies more than 10 percent from design, adjust the drive of each fan to obtain the approximate required cfm. Make a preliminary survey by spot checking air circulation in various areas. Then follow all procedures as outlined after the exhaust system is balanced.
5. Make pitot tube traverse on all main exhaust ducts to determine the air distribution. Investigate any branch that is very low in capacity to make sure that no blockage exists.

6. Adjust the volume dampers in the main ducts to the appropriate air flow (cfm) requirement.
7. Without adjusting any terminal device, measure and record the air flow at each terminal in the system. Study any radical conditions and correct them. Plan the sequence of branch balancing. In making the adjustments, it is preferable to adjust the branch dampers rather than the dampers at the air terminals. If the throttling process at a terminal damper involves closing the damper to a degree that generates noise, evaluate the design cfm capacity of the branch duct.
8. Working from the branch with the highest measured capacity to the branch with the lowest measured capacity, make adjustments in each branch. Beginning with the inlet device most distant from the branch and proceeding toward the branch connection, make volume adjustments at each terminal as necessary. It is important that the balancer use the proper "k" factor prescribed by the air terminal manufacturer for use in conjunction with a particular instrument. In addition, it is often necessary that the readings at grilles, registers, and diffusers be taken in a position or number of positions prescribed by the manufacturer of the air terminal device.
9. Repeat the branch balancing until the system is in balance.
10. Verify the fan capacity and operating conditions again and make a final adjustment in the fan drive if necessary.
11. Verify the action of all fan shut down controls and air flow safety controls.

Glossary

AMPLITUDE: The maximum displacement from the mean position of oscillation or vibrations.

BTU: British thermal unit; the amount of heat required to raise the temperature of 1 pound of water by 1 degree Fahrenheit.

BUILDING ENVELOPE: The imaginary shape of a building indicating its maximum volume; a transition space where the interaction between outdoor forces and indoor conditions can be watched.

CORROSION: Deterioration of metal by chemical or electrochemical reaction resulting from exposure to weathering, moisture, chemicals, or other agents in the environment in which it is placed.

ELASTIC LIMIT: The greatest stress that a material is capable of sustaining without permanent deformation upon complete release of the stress.

EXFILTRATION: The outward flow of air through a wall, joints, etc.

EXHAUST VENTILATION: The removal of foul air from a space by a mechanical means, such as a fan. Fresh air is allowed to enter through available or controlled openings.

HEAT EXCHANGER: A device designed to transfer heat between two physically separated fluids. It generally consists of a cylindrical shell with longitudinal tubes; one fluid flows on the inside, the other on the outside.

INFILTRATION: The seepage or flow of air into a room or space through cracks around windows, under doors, etc.

LATENT HEAT: The amount of heat that is absorbed in changing the state of a substance without changing its temperature.

LEEWARD: Situated away from the wind.

MODULUS OF ELASTICITY: In an elastic material that has been subject to strain below its elastic limit, the ratio of the unit stress to the corresponding unit strain.

MOMENT OF INERTIA: Of a body around an axis, the sum of the products obtained by multiplying each element of mass by the square of its distance from the axis.

NATURAL VENTILATION: Ventilation by air movement caused by natural forces, rather than by fans.

PERMEABILITY: The property of a porous material that permits the passage of water vapor through it.

PICKUP LOAD: The abnormal rate of heat consumption that takes place when a heating system is first turned on. It represents the heat dissipated in bringing the piping and radiators to their normal operating temperature.

POROSITY: A ratio, usually expressed as a percentage of the volume of voids in a material to the total volume of the material, including the voids. The voids permit gases or liquids to pass through the material.

SENSIBLE HEAT: Heat that changes the temperature of a material without a change in state, such as that which would lead to increased moisture content.

STACK EFFECT: The tendency of air in a shaft or other vertical passage to rise when heated, owing to its lower density compared with that of the surrounding air.

STATIC PRESSURE: The pressure that the fan must supply to overcome the resistance to air flow through the system ductwork and system components.

THERMAL LOAD: A load on a structure that is induced by changes in temperature.

THERMAL SHOCK: The sudden stress produced in a material as a result of a sudden temperature change.

THERMAL TRANSMITTANCE: The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

VAPOR BARRIER: A moisture-impervious layer or coating (such as special paint, or a membrane on roofing felt or on building paper) that prevents the passage of moisture or vapor into a material or structure.

VAPOR PRESSURE: The component of the total pressure that is caused by the presence of a vapor, as for example, by the presence of water vapor in air.

VENTILATION: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

WINDWARD: Situated toward the direction from which the wind is blowing.

Bibliography

American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), *Cooling and Heating Load Calculation Manual,* ASHRAE, Atlanta, GA, 1979.

ASHRAE, *Fundamentals Handbook,* 1997.

ASHRAE, *HVAC Applications Handbook,* 1995.

ASHRAE, *HVAC Systems and Equipment Handbook*, 1996.

ASHRAE, *Standard 62-1989, Ventilation for Acceptable Indoor Air Quality,* 1989.

Baturin, V. V., *Fundamentals of Industrial Ventilation*, Pergamon Press, 1972.

Bevirt, W. David, *Environmental Systems Technology*, National Environmental Balancing Bureau (NEBB), Vienna, VA, 1986.

Bradshaw, Vaughn, *Building Control Systems, 2nd Ed.*, John Wiley & Sons, New York, NY, 1993.

Committee on Industrial Ventilation, *Industrial Ventilation, 23rd Ed.*, American Conference of Governmental Industrial Hygienists, Cincinnati, OH, 1998.

Harris, Cyril, *Dictionary of Architecture and Construction*, McGraw Hill, Inc., New York, NY, 1975.

National Air Filtration Association (NAFA), *NAFA Guide to Air Filtration,* NAFA, Washington, DC, 1993.

NEBB, *Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems, 4th Ed.*, 1983.

Sheet Metal and Air Conditioning Contractors National Association, Inc., (SMACNA) *HVAC Duct Construction Standards Metal and Flexible*, 2nd *Ed.*, SMACNA, Vienna, VA, 1995.

SMACNA, *Manual for the Balancing and Adjustment of Air Distribution Systems*, 1981.

SMACNA, *HVAC Systems Duct Design, 3rd Ed.*, 1990.

Stein, Reynolds, and McGuinness, *Mechanical and Electrical Equipment for Buildings*, 7th Ed., (John Wiley & Sons, Inc., 1986).

Appendix E: Hydronic Systems

Principles, Applications, and Acceptance Testing

Dahtzen Chu
Energy Branch, Facilities Division
U.S. Army Construction Engineering Research Laboratory

and

Charles L. Burton
Leland V. Speirs
Department of Architectural Engineering and Construction Science
Kansas State University

Contents

1 Introduction

The science of heating and cooling with water is known as hydronics. Through years of advancement in technology, it is believed that water is still the most practical, economical, and ecologically safe heat transfer medium. The term hydronics should not be confused with hydraulics, which is the study of fluids in motion or at rest. The scope of this presentation on hydronics is to discuss heating and cooling systems, utilizing water, with their various components, maintenance, and testing, adjusting, and balancing (TAB).

Today's hydronic system can best be thought of as a heat transfer machine, large or small, where a heat transfer medium is used to carry heat to or from an area in accordance with the controls installed. Depending on the needs of the occupants, structure, and codes, the versatility of the hydronic system is limited only to the imagination of the person designing the system. A few basic fundamentals need to be learned before a person can do virtually anything desired in the heat transfer ability of a circulating water or hydronic system.

Whether the system is in a large multistory building project supplied by a central mechanical plant, or a small residence, the same basic fundamentals will still apply.

Purpose of Heating and Cooling Systems

The most basic objective of any heating and/or cooling system is to provide occupants with comfortable spaces so they may live, work, and perform well.

Normal body temperature is 98.6 °F. Food and other items we eat or take into our bodies is converted into energy in the form of heat that maintains the body's temperature, so this heat must be dissipated or taken away. With a body temperature of 98.6 °F, it is well noted that a comfort heating system does not warm us up. All it does is adjust inside conditions so the rate of body heat dissipation makes a person feel comfortable.

If heat is dissipated too fast, occupants feel cool; too slow, they feel hot and perspire. When air temperature and humidity are so high the body cannot rid itself of the heat fast enough, steps must be taken to cool and dry the air for comfort.

Bodies lose heat in three ways: radiation, evaporation, and convection. A warm body (human, steel, wood, plastic materials, etc.) loses heat to other surrounding bodies that are at lower temperatures. When standing next to a heated oven or out in the sun with no wind on a hot summer day, a person feels the heat radiated away from those hot objects until it hits the surface of their body and then absorbs it. This is radiation.

Evaporation mainly takes place through breathing but is rapidly increased as perspiration on the skin is introduced by overheating. The evaporation of this moisture on the skin causes a cooling effect because heat is taken away from the skin to change water into vapor.

Convection is the effect of moving air over a body and taking with it the heat at the surface of the body.

Outdoor Conditions

Since each outdoor temperature is indicative of a different rate of heat loss, a heating system must be capable of operating at more than one rate to be effective and efficient. With a given indoor temperature, each outdoor temperature results in a different temperature which controls the rate of heat loss. Balancing the requirements of the various rates of heat loss calls for different rates of heat supply as the outdoor conditions may require.

Basic Hydronic Systems

Heating

A variety of heat sources and heat radiation are used for differing conditions of application. The source of heat may be from a boiler where the combustion of fuel in many forms provides the heat or from a converter wherein the heat of steam is transferred to the water. Both of these systems are shown in Figures E-1 and E-2.

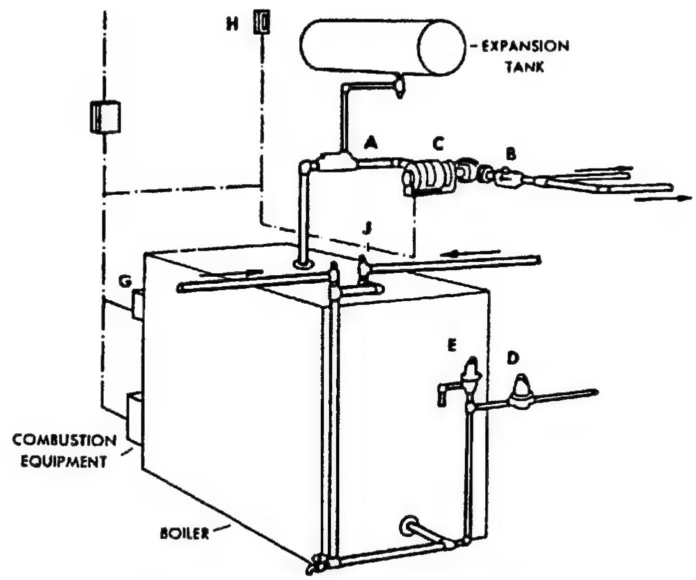


Figure E-1. Heat Supplied by Boiler.
Illustration courtesy of Dunham-Bush, Inc.

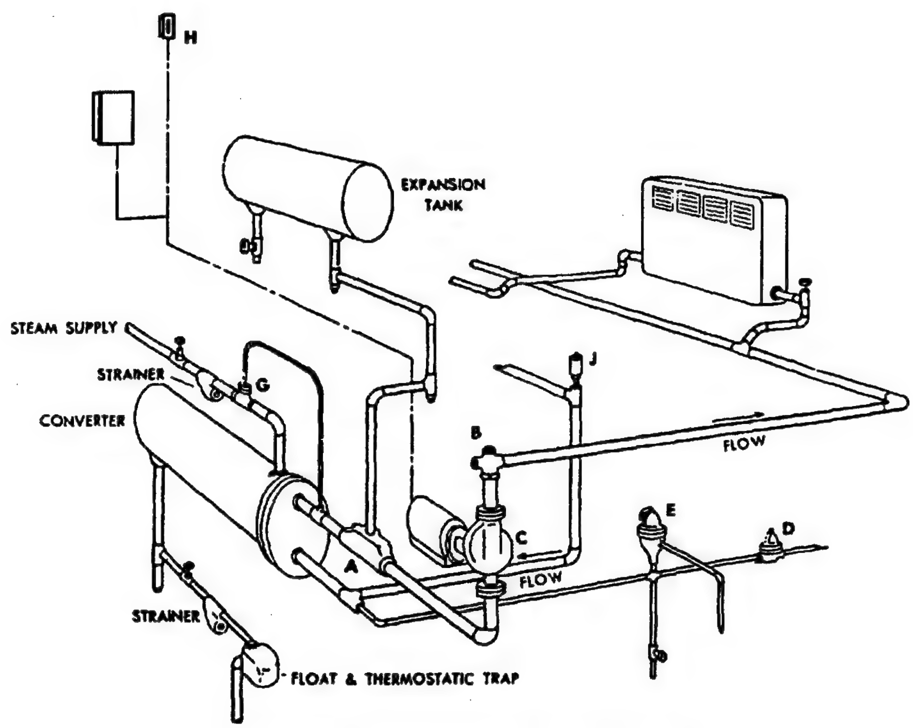


Figure E-2. Heat Supplied by Converter.
Illustration courtesy of Dunham-Bush, Inc.

As described in Figures E-1 and E-2, the main water circuit is from the boiler or converter, through the air separator (A), and flow control valve (B), to the radiation, and back to the boiler or converter.

The flow control valve opens when the pump starts the water's movement and closes when it stops. The closed valve prevents gravity flow of water through the system.

Circulation of water at varying temperatures is controlled in one of several ways to maintain a proper room temperature level. Combustion and pump operations are automatically controlled while a room thermostat is used for control of room temperatures.

Pressure may be raised in hydronic systems to prevent water from steaming so higher temperatures may be obtained, making optional system temperatures obtainable. Where water is used for both heating and cooling, lower water temperatures and open expansion tanks are commonly used.

Cooling

For cooling, a system may be independent of a heating system or combined with one. Such a case of combination would be heating by baseboard or finned pipe units, and cooling by means of a central air-conditioning unit or individual combined heating and cooling units.

Individual fan coil units that operate for heating and cooling may be installed so the same piping is used for both. The arrangement that is more desirable depends on the requirements of the particular installation, including cost considerations. Figure E-3 shows an example of a hydronics heating and cooling system.

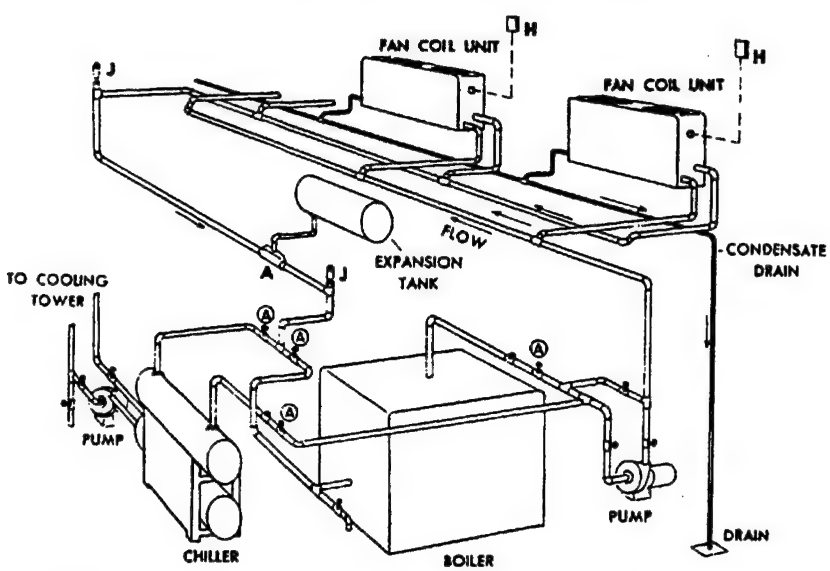


Figure E-3. Hydronics Heating and Cooling System With Change-over Valves.
Illustration courtesy of Dunham-Bush, Inc.

The change-over valves at the locations indicated by a circled "A" permit the system to change from heating to cooling.

2 Air Control in Hydronic Systems

One problem that continues to crop up frequently in hydronic systems, unless handled properly, is the quantity of air permitted to circulate or clog a system. Air control systems that are properly designed and installed can eliminate major problems, reduce maintenance, cut cost of operation, and perform efficiently. Effective air control will also prolong the life of the system and reduce unnecessary noise. Depending on the type of system used, the admission of air to the system will vary.

Once-Through and Recirculating Systems

A once-through system passes water through the equipment only once, then discharges it to a sewer. In a recirculating system, water is not discharged, but flows in a repeating circuit from the heat exchanger to the refrigeration equipment and back to the heat exchanger. Both of these types are further classified as open or closed systems.

Open and Closed Systems

An open system is one in which the water flows into a reservoir open to the atmosphere (i.e., cooling towers and air washers).

A closed system is one in which the flow of water is not exposed to the atmosphere at any point. These systems usually contain an expansion tank that is open to the atmosphere, but the water area exposed is insignificant.

Open systems are piping circuits pumped or gravity circulated. Closed systems are designed and installed as hermetically sealed systems. Some advantages of the closed or "sealed" system follow:

- When a system is closed, little if any make-up water is ever required.

- With no addition of fresh water, there can be no accumulation of oxygen and other corrosive agents. System life is extended indefinitely.

- Closed systems can be pressurized permitting elevated water temperature drops. Piping and operating costs can be reduced significantly.

- Closed systems with positive air control offers improved control, faster temperature response, and quieter system operation.

Whenever possible, closed hydronic systems are used for the reasons given above. Sometimes design conditions require the use of an open system. An example is the use of a water tower when the heat load to be removed from a space or spaces requires greater than 100 tons of refrigeration. The cooling tower allows water to dissipate heat to the atmosphere for these large cooling capacities, thus exposing water to the atmosphere also.

Although a system may be designed as a closed system, too often they end up as an open system unless all components are pressure-tight and leak-proof. Special consideration should be given to pump seals, manual air vents, and tight installations.

Mechanical seals. Mechanical seals are required for all closed system circulating pumps. A specific kind of pump seal, known as the "packing gland" type, requires constant water leakage to provide seal lubrication. This means fresh water must be constantly added to the system or, theoretically, the system will eventually run dry.

Because fresh water contains air and other corrosive agents, system life and operation will be seriously affected. Also, foreign particles of sand and dirt that are often found in fresh water supplies will enter and accumulate within the system. Therefore, water-tight mechanical pump seals should always be used on closed hydronic systems.

Manual air vents. Manual air vents should be used where initial venting of high points in the system is necessary to fill the system with water. Automatic air vents, if allowed to operate automatically after the system is placed into operation, are a source of system leakage. A given amount of air or gas space is required in every system to accommodate water expansion and pressurization.

The first step in providing adequate air control is to design a closed system that is actually closed. Not only must a system be designed as a closed system, but it must also be **installed** as a closed system. Proper installation techniques with adequate allowance for pipe expansion and contraction are necessary.

Unnecessary gauge glasses, particularly when installed in compression tanks, are common sources of air and water leaks. Packing materials in the glass assembly may dry out, and allow air leakage. Normally, gauge glasses on compression tanks serve very little purpose because tanks are often located in high and inaccessible locations. Furthermore, the visible indication of an air level in a tank does not always mean that a proper air cushion is present.

Air levels that vary at different times and in different systems are more apt to confuse than to help. Compression tanks, therefore, should be constructed with a minimum of openings, preferably with no openings located above the water line.

3 Components of Hydronic Systems

Warmed water heating and chilled water cooling systems are at their best levels of performance when the terminal equipment and piping circulation is positive and balanced, free of air, and the systems are under proper pressure.

The devices that perform or promote these functions are called "specialties" and include: air separators, air vents, flow valves, pressure relief valves, pressure reducing valves, radiator valves, balancing fittings, vent tees, and diverter fittings. Small capacity low head circulating pumps and small closed expansion tanks sometimes are also called specialties.

Because the principles of selecting these specialties for systems are a matter for the design engineer to specify, only a brief description of each will be given here.

Component Locations

The system shown in Figure E-4 indicates the location of the before-mentioned specialties.

Air Separator

Number 1 in Figure E-4 indicates an air separator. An air separator (Figure E-5) releases entrained air from the water before it reaches the piping system and radiation-convectors for fan-coil units. It consists of a baffled chamber. The baffle in it is positioned to create the required amount of turbulence and deflect the flow to cause air bubbles to rise and accumulate in the upper part of the separator. From there, it passes up and into the expansion tank to help maintain the air cushion. Air separators range in size from 1 to 4 in.

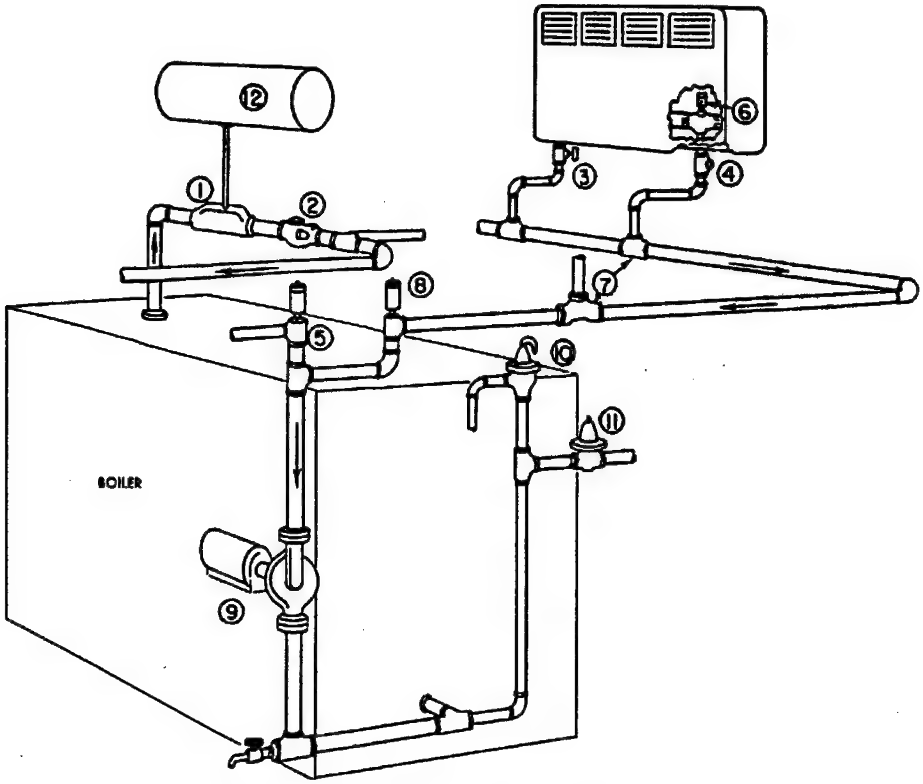


Figure E-4. Hot Water System Specialties.
Illustration courtesy of Dunham-Bush, Inc.

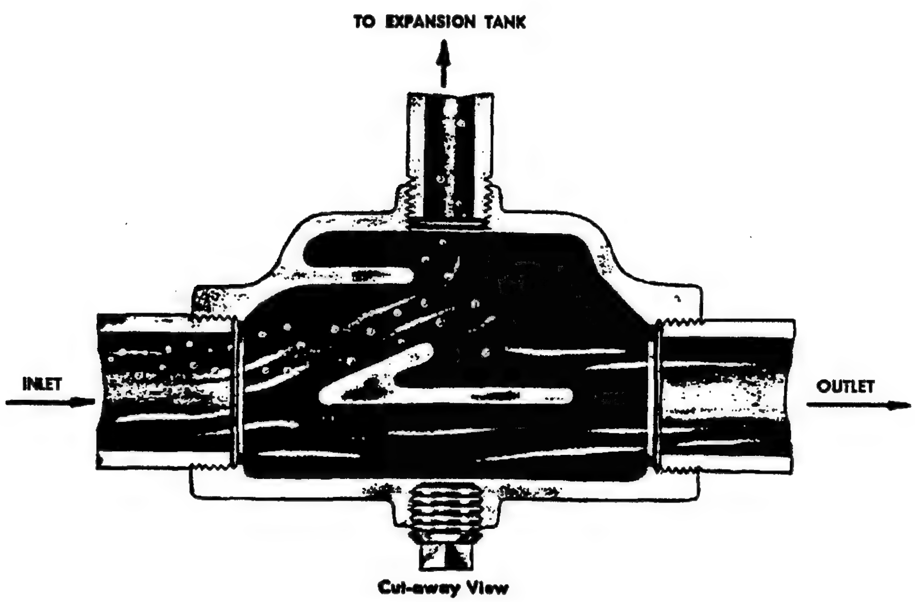


Figure E-5. Air Separator.
Illustration courtesy of Dunham-Bush, Inc.

Air separators should be installed horizontally in the supply main about 18 in. from the point where the main rises vertically from the boiler or converter as shown in Figure E-6.

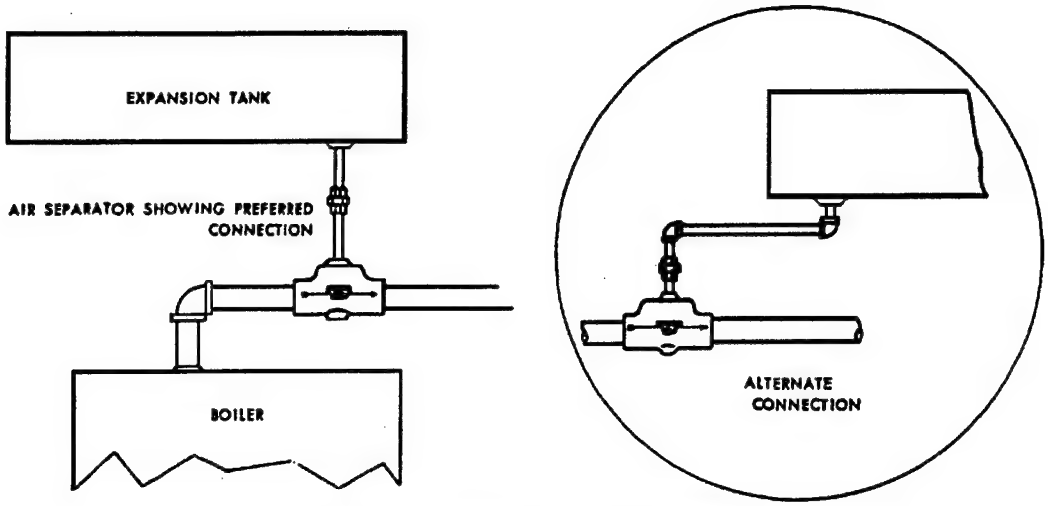
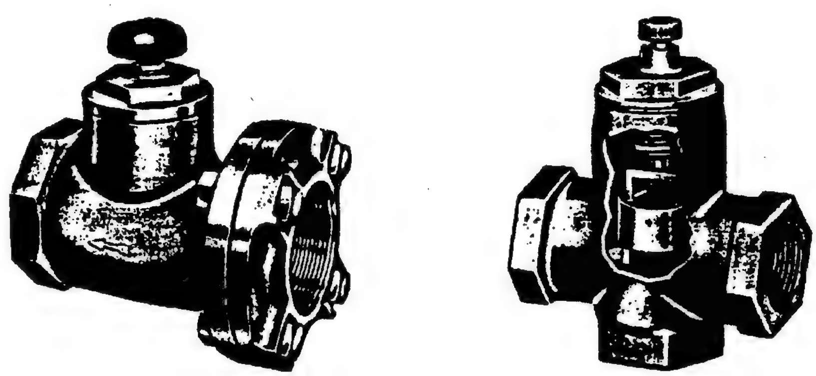


Figure E-6. Installation of Air Separator.
Illustration courtesy of Dunham-Bush, Inc.

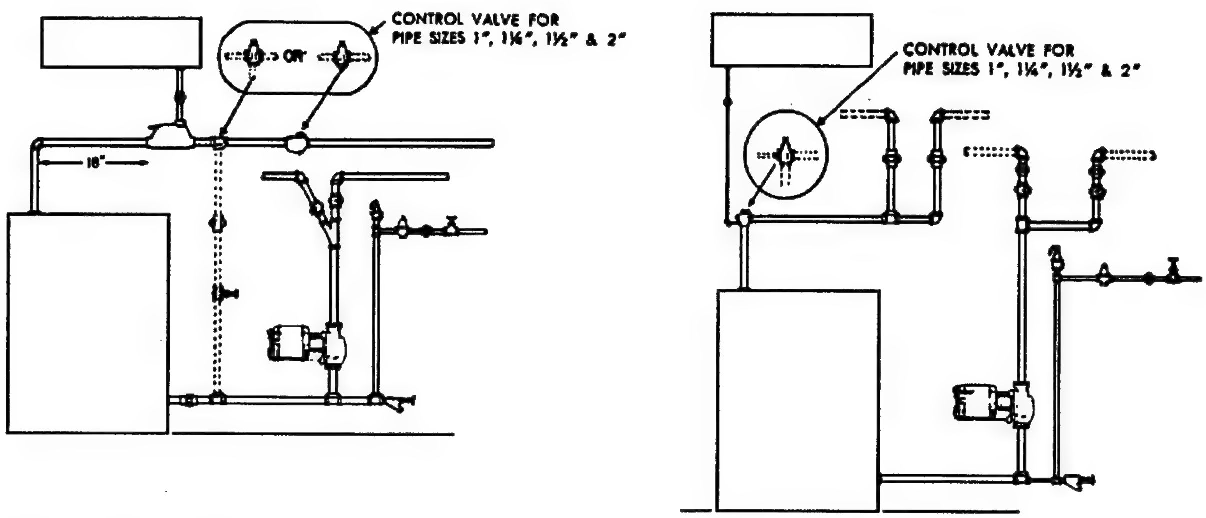
Flow Control Valve

Number 2 in Figure E-4 indicates the proper location of the flow control valve. Flow control valves control the direction of water flow and prevent circulation of hot boiler water when heat is not needed, as when the circulating pump has been stopped by its control. This permits the boiler water to be used at temperatures high enough to heat domestic water in winter and summer since the valves prevent circulation by gravity. These valves are made in two patterns, horizontal (straightway) and universal pattern as in Figure E-7.



(a) Horizontal Pattern (b) Universal Pattern
Figure E-7. Flow Control Valves.
Illustration courtesy of Dunham-Bush, Inc.

The horizontal pattern valves are manufactured in 2-, 2-1/2-, and 3-in. sizes while the universal pattern used for pipe sizes 1 to 2 in. inclusive may be installed for either angle or straightway application. The flow control valve should be installed in the main beyond the air separator, before any branches are taken off (see Figure E-8).



(a) Horizontal Pattern Flow Control Valve With Optional Universal Pattern

(b) Universal Pattern Flow Control Valve

Figure E-8. Flow Control Valves.
Illustration courtesy of Dunham-Bush, Inc.

When the pump is running, the disc on top raises to open the valve; when the circulator stops, the disc closes tight to prevent gravity flow. The external adjustment arm may be positioned to "open," "normal," or "closed" settings. For regular winter or summer operations, it is set at "normal." Every zone should include a flow control valve.

Circulator Valve

Number 3 of Figure E-4 indicates the location of a circulator valve. Terminal equipment such as convectors and fan-coil units should be valved individually. When applied to these terminal units, circulator valves (Figure E-9) give occupants instantaneous on-or-off control of water flow. Large self-cleaning waterways reduce the water resistance and circulator load, and the T-type handles indicate the inner valve position that opens and closes fully in 1/4 turn. These valves may be set for partial flow.

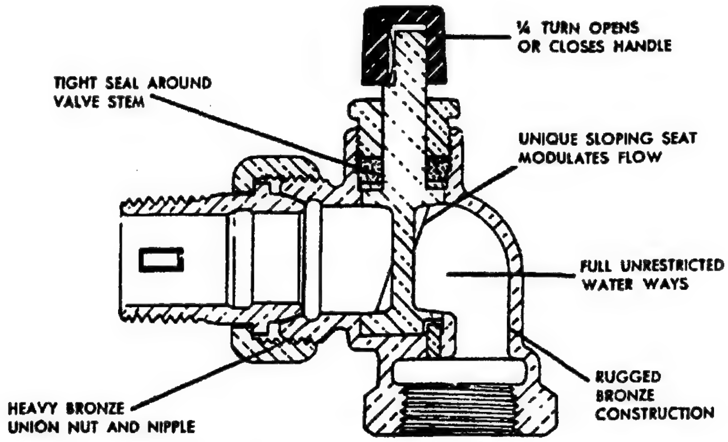


Figure E-9. Circulator Valve.
Illustration courtesy of Dunham-Bush, Inc.

Balancing Elbow and Balancing Fitting

Number 4 of Figure E-4 indicates the location of a balancing elbow and a balancing fitting. The flow balancing that cannot be handled by piping design must be handled by other equipment. Balancing elbows and fittings permit balancing the water to individual terminal equipment.

Once properly adjusted, each radiator or other terminal equipment receives the water flow needed for even heat. Balancing fittings for floor panels should be conveniently accessible for adjustment after the concrete floor has been poured.

Vent Tee

Number 5 of Figure E-4 indicates the location of a vent tee. Vent tees are a convenience for arranging for the application of air vents. They save space for base board and perimeter type convection radiation. A 1/2-in. tapped opening eliminates the need for bushing or drilling and tapping.

Expansion Type Air Vent

Number 6 of Figure E-4 indicates the location of an expansion type air vent. Air vents that release the air automatically promote effectiveness of terminal equipment. The type shown in Figure E-10 operates on the hygroscopic principle that allows the least amount of system leakage.

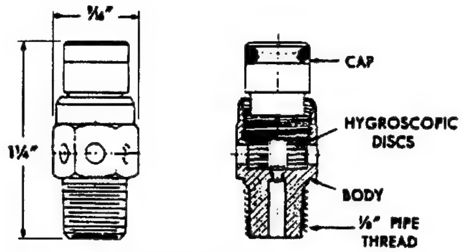


Figure E-10. Expansion Type Air Vent.
Illustration courtesy of Dunham-Bush, Inc.

Composition discs that become wet and expand, seal venting points when the system is filled. Entrapped air dries and contracts the discs, opening the venting ports. The vents require no adjustment.

The vents may be manually operated by turning the cap counter-clockwise 3/4 turn to permit rapid air removal when the system is first filled. Each piece of terminal equipment, as well as high points of mains and branches where air might collect, should be equipped with a vent.

Vents should not be installed inverted; however, they do operate satisfactorily in the vertical upright or horizontal position. Some venting applications are shown in Figure E-11.

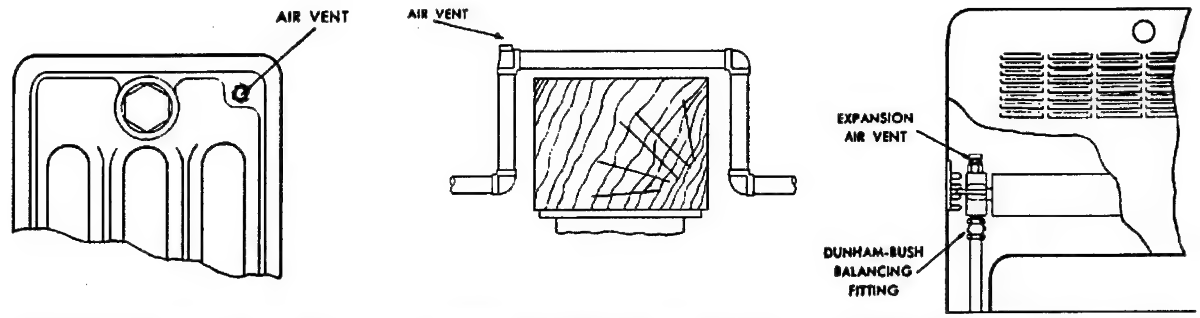
Diverter Fitting

Number 7 of Figure E-4 indicates the location of a flow diverter fitting. Diverter fittings were developed to reduce the labor and inconvenience of using complicated connections at a single main radiator. These fittings proportion the energy in the flowing stream to provide needed energy by imposing resistance to flow in the main.

They are needed on all systems and zones piped for a single pipe circuit, except where the radiation itself serves as a portion of the main—as in baseboard systems.

Float Type Vent

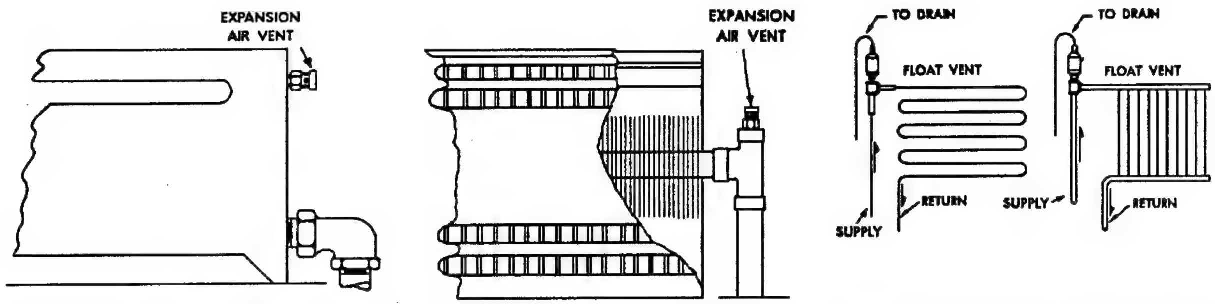
Number 8 of Figure E-4 indicates the location of the float type vent. Some points along the piping system and certain types of radiation require prompt, rapid venting. The high points of downfeed systems and the ends of mains of larger systems are locations where proportionately large venting capacity is needed. Figure E-12 illustrates an example of a float type vent.



Standing Hot Water Radiation Application

Venting High Points Application

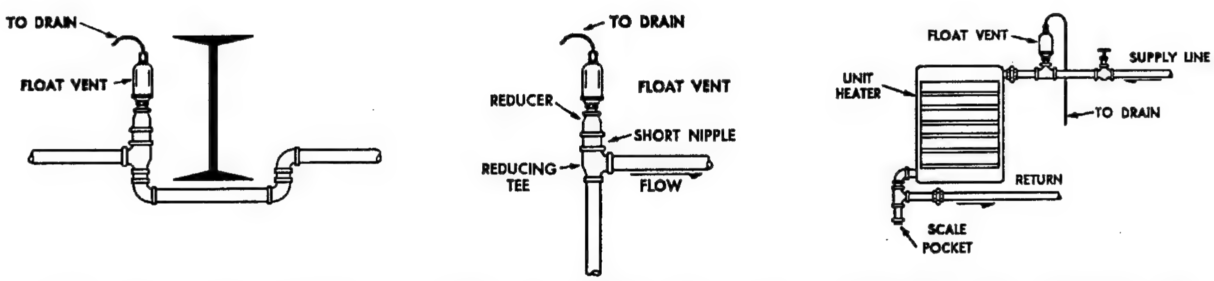
Hot Water Convector Radiation Application



Cast Iron Hot Water Baseboard Application

Hot Water Baseboard Radiation Application

Types of Panel Coils--Wall Application



Trapped Mains or Circulating Pipes Application

High Point on Water Mains Domestic Hot Water Application

Hot Water Unit Heater Application

Figure E-11. Typical Air Vent Installations.

Illustration courtesy of Dunham-Bush, Inc.

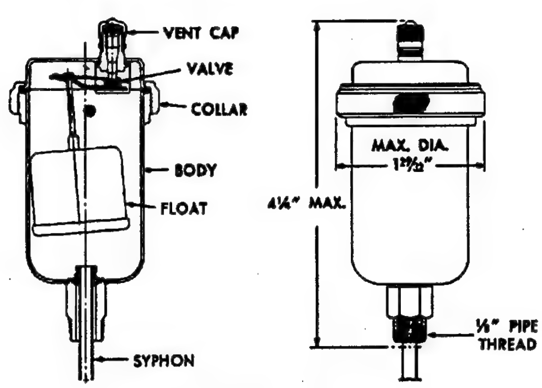


Figure E-12. Float Type Vent.

Illustration courtesy of Dunham-Bush, Inc.

Float type vents operate continuously and automatically, and serve both warm and chilled water lines. Air vents constructed to eliminate water logging, as is the case with the float type vents shown in Figure E-12, are preferable.

They should be installed with drain tubes arranged to discharge to a suitable place whenever their location presents the possibility of damage to the construction or furnishings, if the vent should spurt water.

Hydronic Circulators

Number 9 of Figure E-4 shows hydronic circulators. Please refer to Section 6 (of this appendix) on hydronic pumps and circulators.

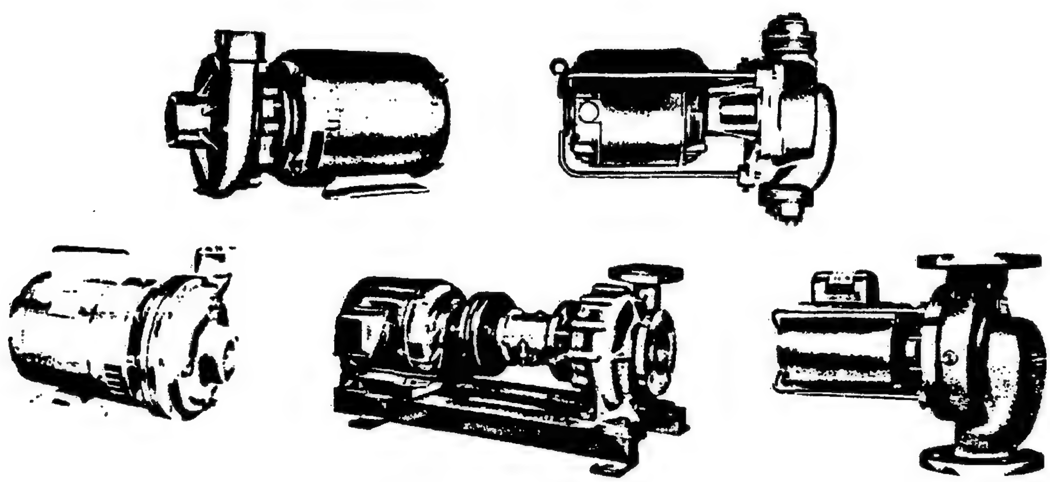


Figure E-13. Circulators.
Illustration courtesy of Dunham-Bush, Inc.

Pressure Relief Valve

Number 10 of Figure E-4 indicates the location of a pressure relief valve. Low-pressure systems are usually limited to 30 psig pressure. Relief valves (Figure E-14) are needed with systems using closed expansion tanks. Some relief valves are equipped with a diaphragm arrangement that transmits more power during opening of the valve; a spring chamber seals an adjusting spring against corrosive action of discharge water. Systems larger than 250,000 Btu/h may require two valves.

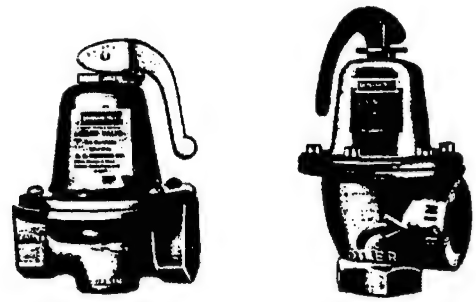


Figure E-14. Pressure Relief Valves.
Illustration courtesy of Dunham-Bush, Inc.

Pressure Reducing Valve

Number 11 of Figure E-4 indicates the location of a pressure reducing valve. Pressure reducing valves are needed to keep systems with closed expansion tanks under proper pressure and automatically filled. The valves shown in Figure E-15 maintain a minimum water pressure of 12 psig on the system. Any drop of pressure below 12 psig causes the valve to open and feed water into the system.

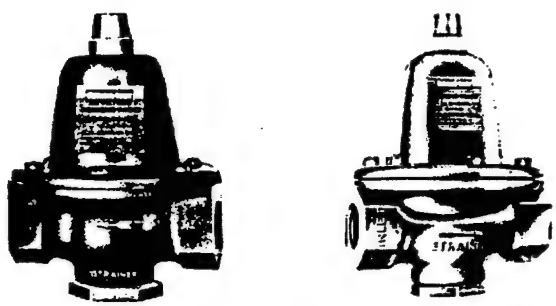


Figure E-15. Pressure Reducing Valves.
Illustration courtesy of Dunham-Bush, Inc.

A common practice for pressure reducing valves and relief valves is to install them in tandem. For this type of application, dual units, as illustrated in Figure E-16, are convenient.

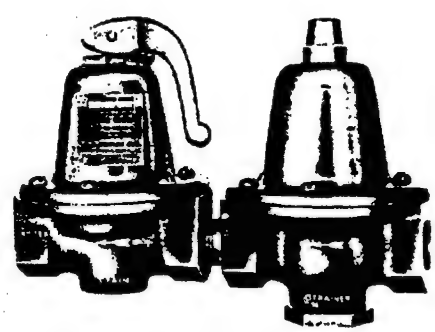
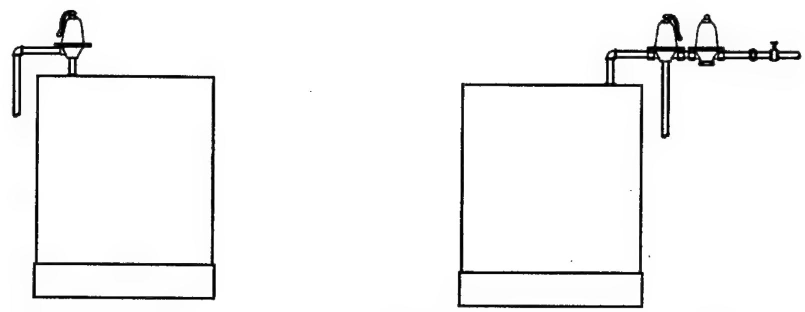


Figure E-16. Dual Unit Valves.
Illustration courtesy of Dunham-Bush, Inc.

Pressure reducing valves and dual unit valves are installed as indicated in Figure E-17.



(a) Relief Valve (b) Dual Unit Relief and Reducing Valve

Figure E-17. Relief Valves.

Illustration courtesy of Dunham-Bush, Inc.

Expansion Tank

Number 12 as illustrated in Figure E-4 indicates the location of an expansion tank.

4 Piping Methods

An important contribution to satisfactory performance is the method of piping one selects in the design process. Some concerns in the selection of the type or method chosen are arrangement, construction, occupancy and size of the building, and space available for the piping. Low first cost is important, but more importantly, the piping must promote low overall system cost with economy, reliability, and ease of operation. With each piping system comes inherent flow characteristics and corresponding adaptability for particular purposes.

What each of these factors can provide for satisfactory system performance is the basis for selecting the piping system to be used. Sometimes the necessity for keeping first cost at a minimum determines whether the piping shall be arranged so that it tends to be "self-balancing" or whether balancing devices will be used periodically. The deciding criterion is the cost of the additional material needed to pipe the system so it tends to be "self-balancing" compared to the cost of balancing after installing it. In small systems, balancing the flow after installation is usually only a minor problem, so the savings in material makes for the lowest installed cost.

Hydronic distribution supply and return mains are often located at corridor ceilings, above hung ceilings, wall-hung along a perimeter wall, and in pipe trenches, crawl spaces, or basements. System piping does not need to be run at a definite level or pitch.

Water System Piping Classification

Water system piping can be divided into two classifications: circuits for small systems and main distribution piping.

Circuits for Small Systems

These are pipe circuits suitable for complete small systems or as terminal or branch circuits on large systems.

- Series loop
- One-pipe
- Two-pipe reverse-return
- Two-pipe direct-return.

Main Distribution Piping

Main distribution piping is used to convey water to and from the terminal units or circuits in large system.

- Two-pipe direct-return
- Two-pipe reverse-return
- Three-pipe
- Four-pipe.

Pipe Circuit Types

Series Loop System

A series loop system is a continuous length of pipe or tube from a boiler or chiller supply connection back to the boiler or chiller return connection. Terminal units are part of the loop.

One or many series loops may be used in a complete system. For example, a multistory building may have a series loop on each floor with supply and return to the same source. While giving off heat, the water temperature drops continually as each room terminal unit transfers heat to the air. The amount the temperature drops will depend on the water flow rate and unit output. Comfort cannot be maintained in separate spaces heated with a single series loop if water flow rate is varied. This is due to the following simple facts: the average water temperature shifts down progressively from the first to last unit in series, and unit output gradually lowers from first to last on the loop. Figure E-18 is an example of a series loop system with baseboard forced-circulation warm water installation.

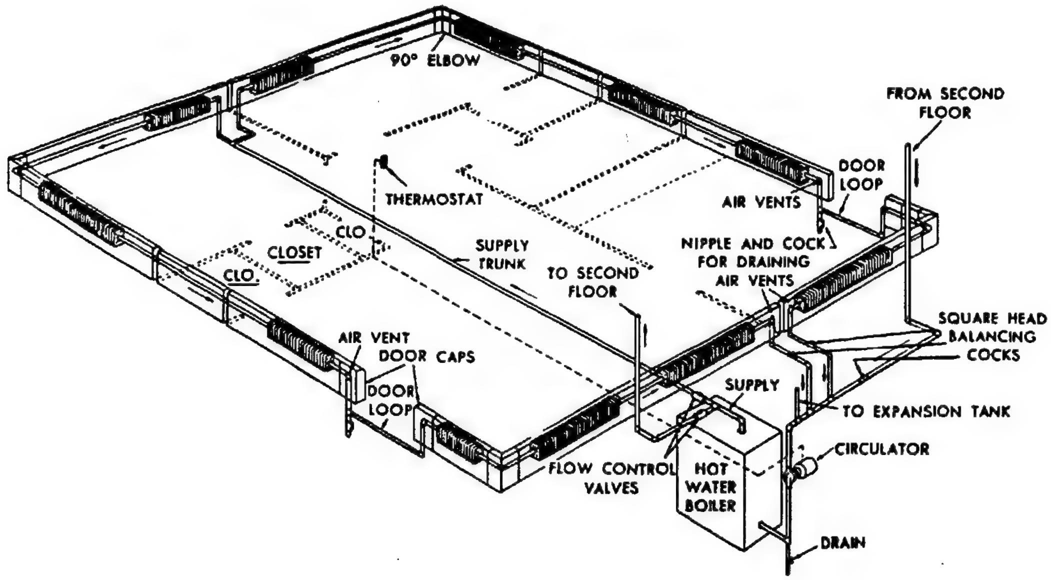


Figure E-18. Series Loop System.
Illustration courtesy of Dunham-Bush, Inc.

One-Pipe Systems

One-pipe circuits make use of a single loop as a supply and return main. For each terminal unit, a supply and a return tee are installed on the same main. One of the tees is a diverting tee that creates a pressure drop in the main flow to divert a portion of the main flow to the unit.

Unlike series loop systems, one-pipe circuits allow manual or automatic control of flow to individually connected heating units. An on-off rather than flow modulation control is recommended because of the relatively low pressure and flow diverted. A one-pipe system is shown in Figure E-19.

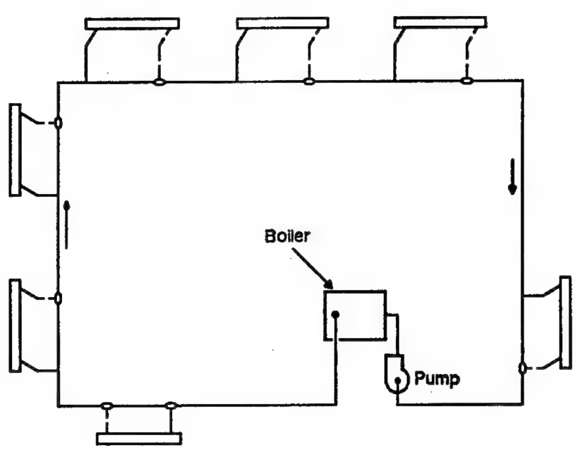


Figure E-19. One-Pipe System.
Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Two-Pipe Systems

Two-pipe circuits may be direct-return or reverse-return. In the direct-return, the return main flow direction is opposite supply main flow; return water from each unit takes the shortest path back to the boiler. This is indicated in Figure E-20.

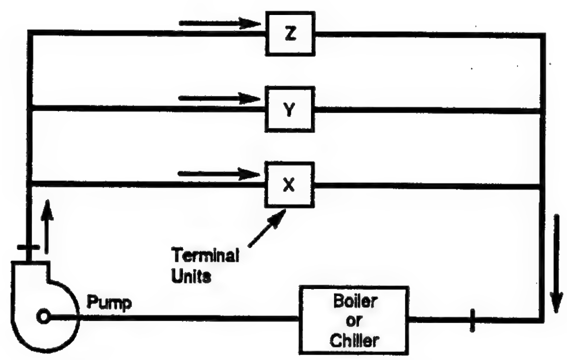


Figure E-20. Two-Pipe Direct-Return.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

In the reverse-return, the return main flow is in the same direction as supply flow; after the last unit is fed, the return main returns all water to the boiler (Figure E-21).

The direct-return system requires less return main piping, but circuit valves are usually required on units or subcircuits. Because water flow distance to and from the boiler is nearly the same through any unit on a reverse-return system, balancing valves require less adjusting. Pumping costs are likely to be higher in the direct-return system because of the added balancing fitting drops at the same flow rates.

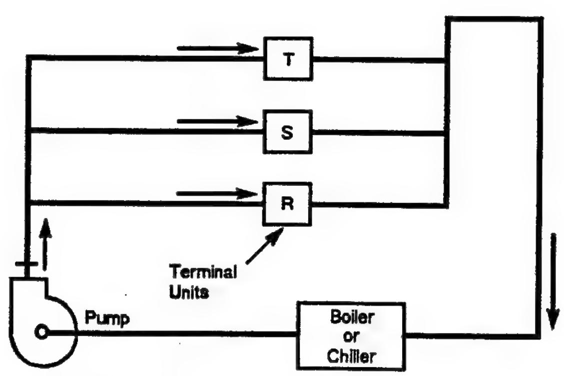


Figure E-21. Two-Pipe Reverse-Return.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Three-Pipe Systems

A three-pipe system (Figure E-22) is usually used with an induction system and will satisfy the variation in load by providing independent sources of heating and cooling to the room unit in the form of constant temperature primary or secondary chilled and hot water. If used with an induction unit, it contains a single secondary water coil.

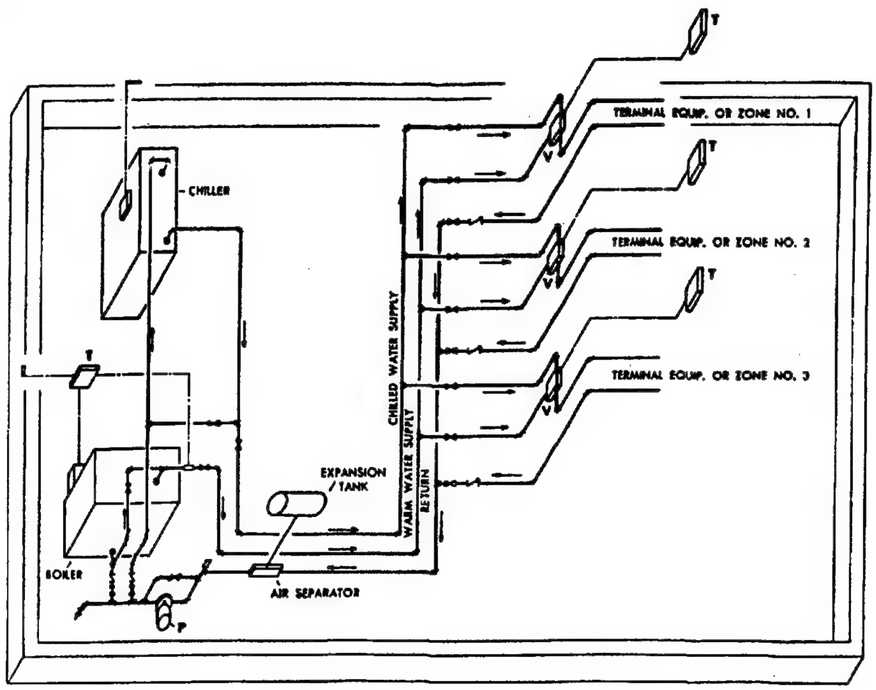


Figure E-22. Three-Pipe System.

Illustration courtesy of Dunham-Bush, Inc.

A three-way valve at the inlet of the coil admits the water from the hot or cold water supply as required. The water leaving the coil is carried in a common pipe to either the secondary cooling or heating equipment.

Four-Pipe Systems

Systems for induction and radiant panel or fan-coil systems derive the name four-pipe systems because of the four pipes to each terminal unit. The four pipes consist of a cold water supply, a cold water return, a warm water supply, and a warm water return.

The four-pipe system (Figure E-23) satisfies variations in cooling and heating to the induction units using temperature primary air, secondary chilled water, and

secondary hot water. Terminal units are provided with two independent secondary water coils; one served by hot water and the other by cold water.

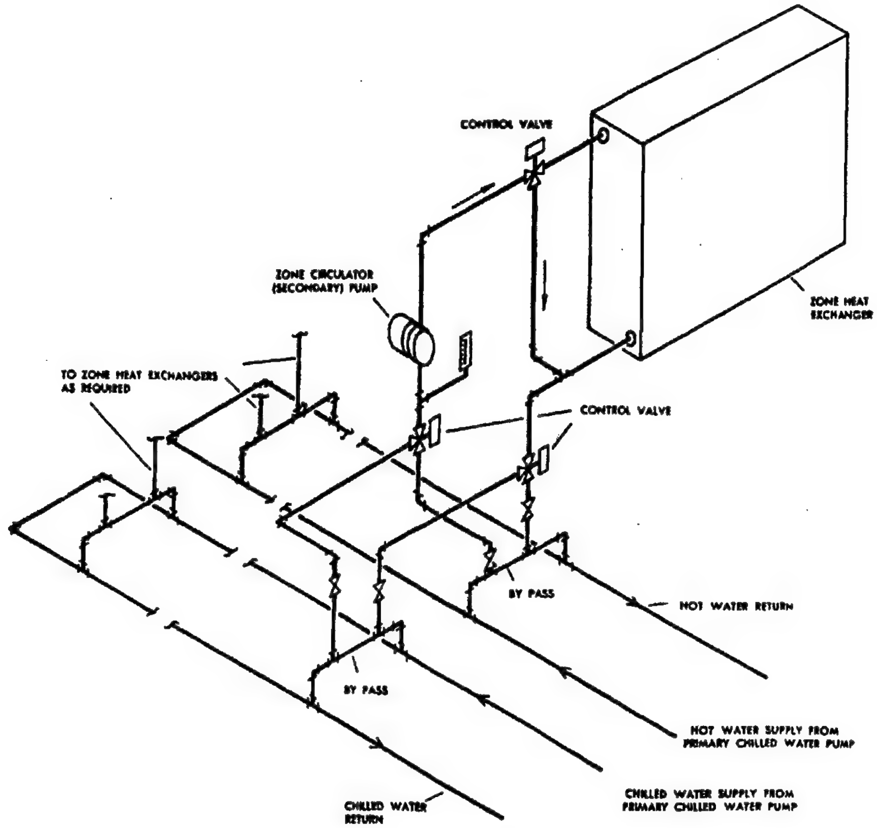


Figure E-23. Four-Pipe System.

Illustration courtesy of Dunham-Bush, Inc.

5 Terminal Equipment

The equipment used in conditioning space air in hydronic comfort cooling, air-conditioning systems, and space heating equipment is referred to as terminal equipment. Space heating equipment, however, is more frequently referred to as "radiation."

Terminal equipment used for cooling includes unit room coolers, unit space coolers, and air-handling units.

Terminal equipment used in heating includes converters, storage water heaters, heat exchange equipment at which supply pipes terminate, free-standing radiation, convectors, finned tube radiation, unit heaters, unit ventilators, and central fan heaters.

Cooling Equipment

Conditioning air, at a minimum, would include simultaneous control of temperature, humidity, motion, and cleanliness of air. Satisfactory conditioning of air depends not only on equipment selection but also on applying it so the following conditions are met:

- Uniform air temperatures each space requires.
- Comfortable humidity or wet bulb temperature required.
- Continuous, mild, uniform air motion within the occupied level; no drafts of contrasting air movement, such as a variable speed fan may cause.
- Proper control of mean radiant temperature.
- Freedom from odors by regulating air changes per hour.
- Air cleanliness prescribed by code or occupancy.

- Noise criterion.

- Maintenance and operating procedures.

Unit Room Coolers

Unit room coolers are sometimes referred to as fan-coil units and are capable of cooling, heating, or both. This type of conditioning equipment uses chilled water as its cooling medium and is usually located a good distance from the chiller. Sometimes it is referred to as "remote conditioning equipment." Several types of fan-coil units are used today.

Vertical units.

- Cabinet: This type is located within the space cooled (Figure E-24).

- Basic: These are used for concealment within the building structure or in a cabinet and enclosure housing the piping or finned radiation (Figure E-25).

- Recessed: Made for recessing in the building's wall (Figure E-26).

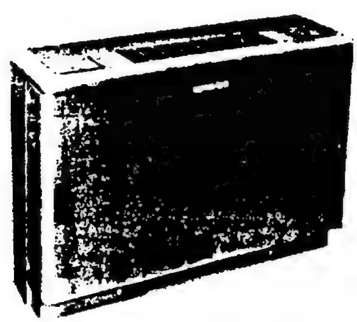


Figure E-24. Cabinet Type Fan-Coil Unit.

Illustration courtesy of Dunham-Bush, Inc.

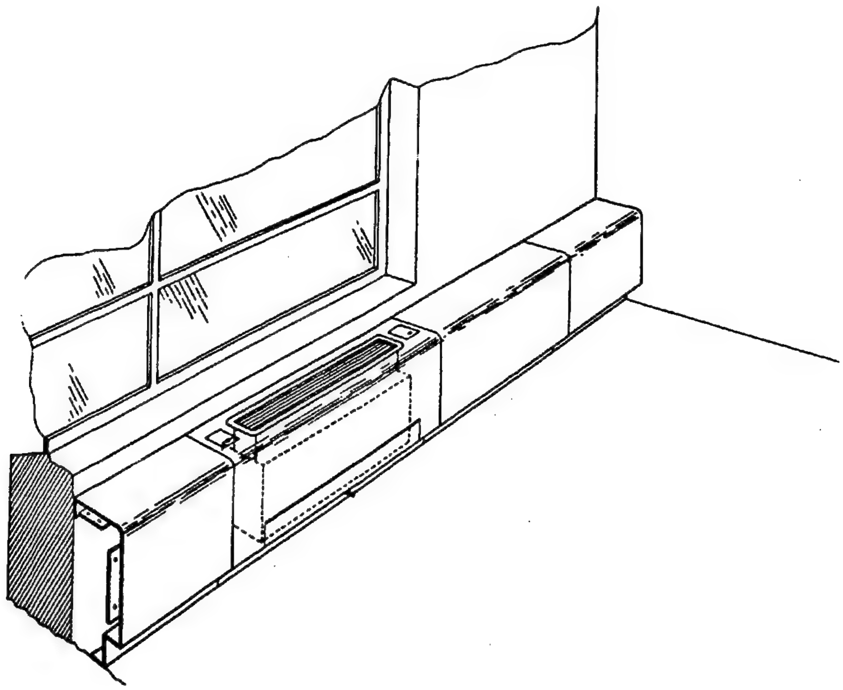


Figure E-25. Basic Type Fan-Coil Unit.

Illustration courtesy of Dunham-Bush, Inc.

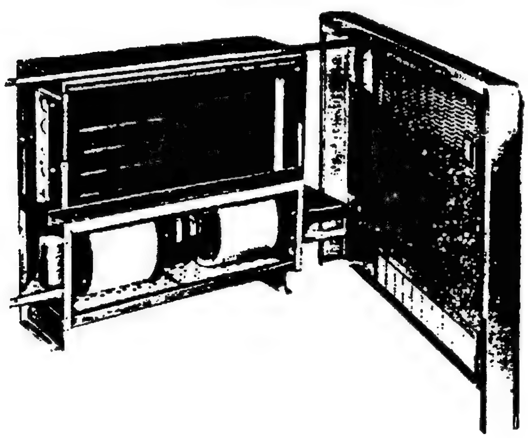


Figure E-26. Recessed Type Fan-Coil Unit.

Illustration courtesy of Dunham-Bush, Inc.

To better visualize the components of the cabinet, recessed, and basic fan-coil unit, see Figure E-27.

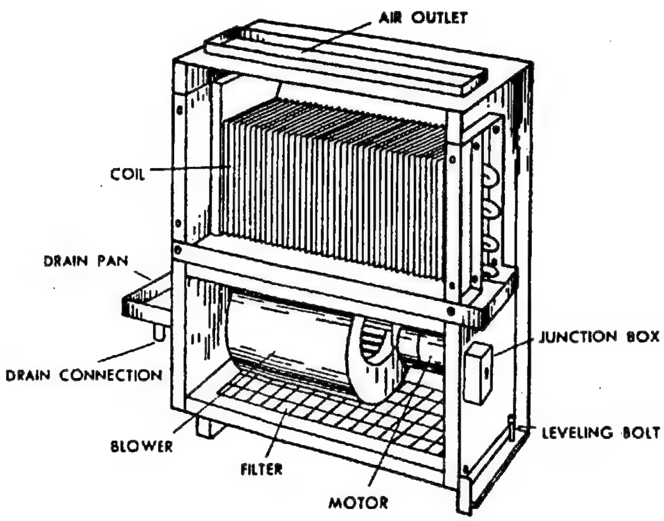


Figure E-27. Components of a Fan-Coil Unit.

Illustration courtesy of Dunham-Bush, Inc.

Horizontal units.

Cabinet: These units are hung from the ceiling of the conditioned space. An insulated case encloses the blower overhung on the motor shaft, drain pan with connection, and space for the control and valve equipment. Figure E-28 illustrates this concept.

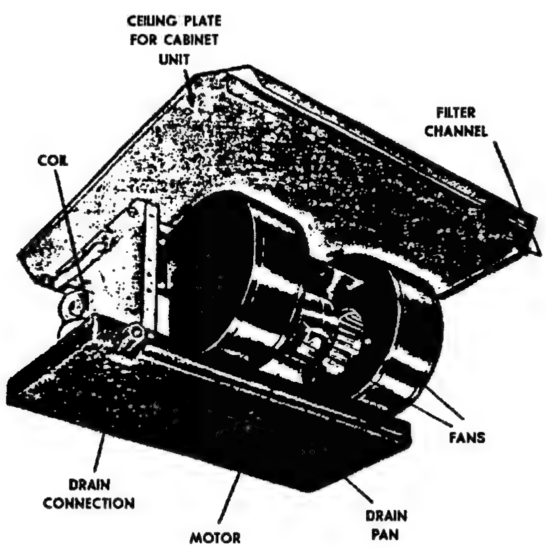


Figure E-28. Cabinet Type Horizontal Unit.

Illustration courtesy of Dunham-Bush, Inc.

Basic: The basic unit is mounted in the ceiling of the space. The case is formed by the enclosed building construction, but the construction is otherwise the same as for the cabinet unit. Figure E-29 shows a sample ceiling construction and unit.

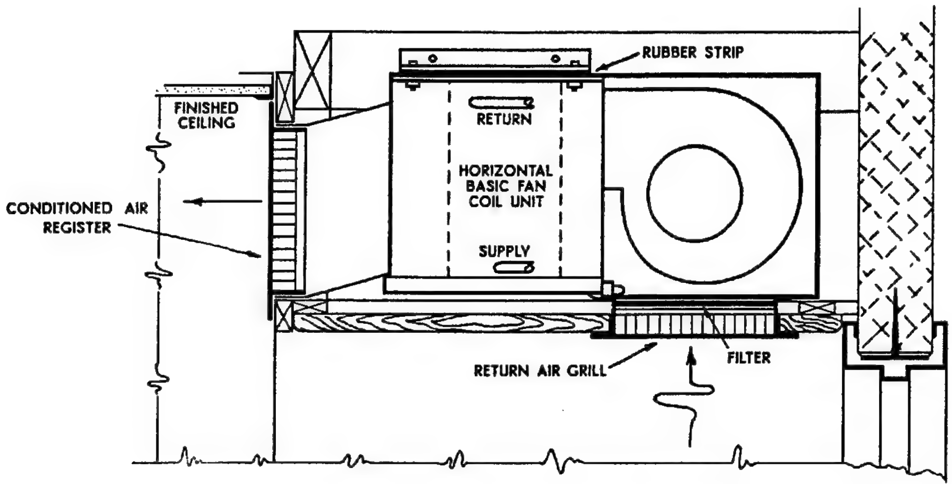


Figure E-29. Basic Horizontal Fan-Coil Unit.

Illustration courtesy of Dunham-Bush, Inc.

Unit Space Coolers

Unit space coolers are the same as unit room coolers, but differ in their capacity. Because of this difference, their construction is slightly different. They use a single fan, double width, and double inlet for all sizes. They are capable of operating against a higher static pressure than room units, including the pressure drop within the unit through filter and coil. The throw is greater for a free outlet discharge than for the horizontal units. Ventilation air may be supplied through-the-wall by use of mixing boxes that are built on the job by the installer.

Air-Handling Units

Air-handling units are usually assembled in the factory and are much the same as a central fan or conditioning system. Figure E-30 shows several types of air handling units.

An air-handling unit may be used to lift a load from an interior space that has no heat loss but only gains from equipment, people, and lighting. It may also be used where the air from separate rooms may not be mixed.

These units are typically used for cooling, heating, or both, wherever large capacities and correspondingly long throws are required. They are also used when specific air conditions are to be provided.

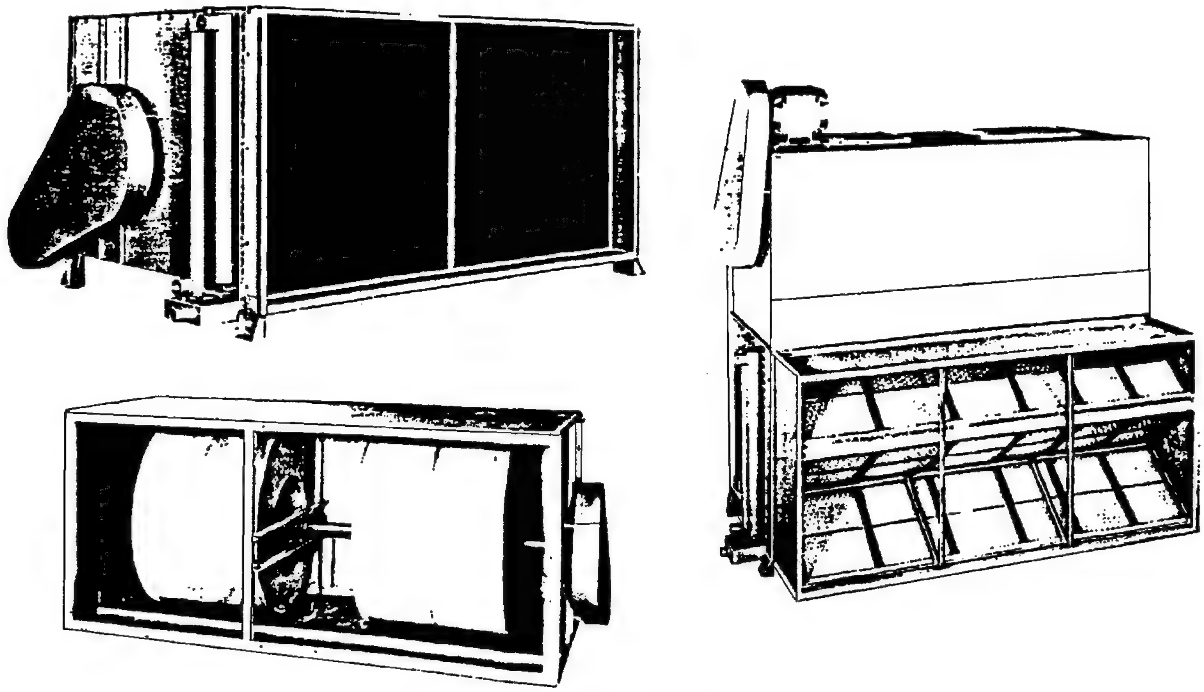


Figure E-30. Air-Handling Units.

Illustration courtesy of Dunham-Bush, Inc.

Heating Equipment

Some general guides for applying terminal equipment for heating are as follows:

- Locate enough radiation in bays and alcoves to avoid pockets of cold air in these particular areas.
- In long rooms, space the radiation at proportionate distances from each other so the radiation will spread uniformly through the occupancy.
- Enough extra radiation should be supplied to rooms (such as kitchens, bathrooms, etc.) that have exhaust fans to compensate for heat being lost through the exhaust.
- Radiation should be located under windows and along perimeter walls of the building. These locations have down drafts of cold air which need to be counteracted.
- On landings in a stairway, radiation will counter down drafts that occur in these areas.

Perimeter Radiation

When exterior walls are constructed in such a way that cold surfaces can result on the inside of the wall, uniform temperatures are not attainable unless the radiation extends over the exposed wall area or under areas such as glass. For this purpose, finned pipe radiation is used.

Shielding, cleanliness, and appearance are important factors in a unit radiating heat. If dirt and lint particles cover the finned parts of the tube, the heat transfer will be reduced significantly.

Baseboard radiation. The baseboard is a form of convection equipment using finned tubing. The tubing, fins, and shield are placed where the floorboard or baseboard is located around the perimeter of the room. The tubing is usually steel with steel fins or copper with aluminum fins. A finned tube baseboard is shown in Figure E-31.

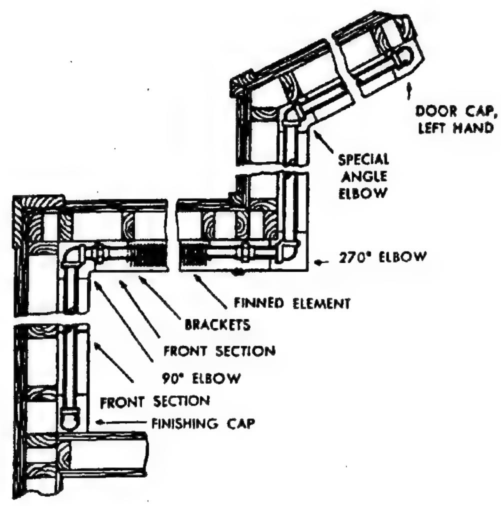
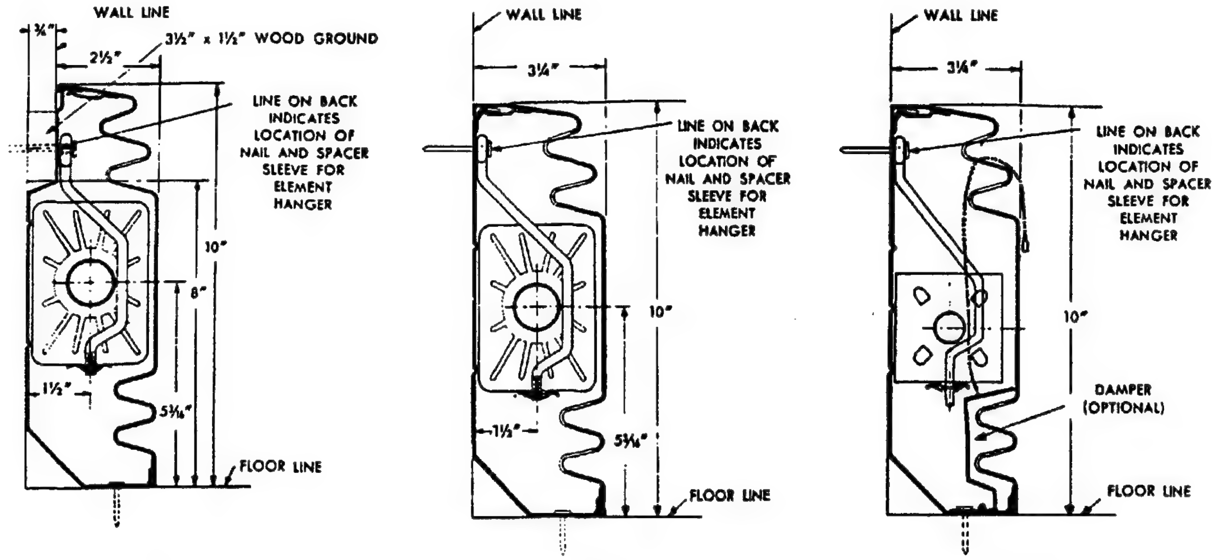


Figure E-31. Finned Tube Baseboard.
Illustration courtesy of Dunham-Bush, Inc.

Systems using finned type radiation and baseboard should be zoned properly. If not zone controlled, piping circuits should be divided so that each supplies portions of the building subjected to the same variable conditions, hours of use, sun, prevailing wind, and so forth.

The three types of standard baseboards—semi-recessed, flush (1-in. tube with 2-3/4-in. x 4-in. fin), and flush (3/4-in. tube with 2-3/4-in. square fin)—are illustrated in Figure E-32.



Semi-recessed
1-in. tube
2-3/4 in. x 4 in. fin

Flush
1-in. tube
2-3/4 in. x 4 in. fin

Flush
1 in. tube
2-3/4 in. square fin

Figure E-32. Standard Baseboard Types.

Illustration courtesy of Dunham-Bush, Inc.

Along-the-wall radiation. The difference between baseboard and along-the-wall radiation is that higher heating capacities are required for industrial, institutional, and commercial buildings. The along-the-wall type radiation has higher heating capacities than finned tube radiation; therefore, larger heating loads may be handled with low height enclosures. An example of along-the-wall radiation is shown in Figure E-33.

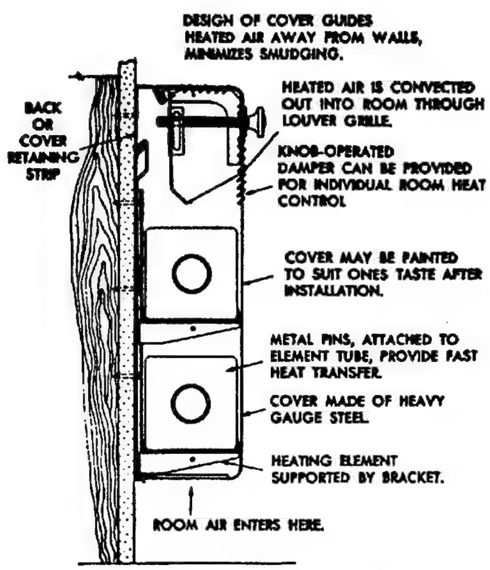


Figure E-33. Along-the-Wall Radiation.

Illustration courtesy of Dunham-Bush, Inc.

Heating Coils

Heat exchangers used for heating air are called heating coils. Heating coils can be used on gravity and mechanically circulated applications. Gravity convection is experienced by the natural flow of hot air rising and cold air falling, while forced convection includes the use of a fan to circulate air.

Coils are used as a primary source of heat for warming the air. They may be located in branch ducts. In this case, they are called booster coil heaters. Coils may also be used to reheat chilled air for humidity control.

Blower Unit Heaters

Like other unit heaters, blower units are fan and heat exchanger coil assemblies in properly braced casings. Blower units are characterized by use of centrifugal fans, which are capable of handling large quantities of air against substantial pressures.

When they are used to heat and circulate air only, they are referred to as unit heaters. The unit may also be equipped to clean or temper the air by using filters or face-and-bypass dampers.

Blower unit heaters (Figure E-34) can be built as vertical or horizontal units. These assemblies are often located a distance from the spaces to be occupied, and supply air is transported to the rooms by ducts.

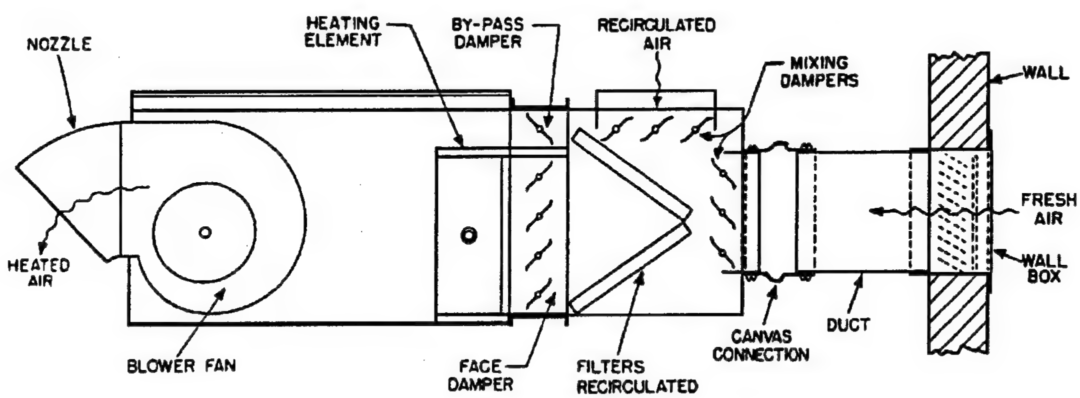


Figure E-34. Blower Unit Heater.

Illustration courtesy of Dunham-Bush, Inc.

Convectors

Convector type radiation supplies heat by slow air movement and mild radiant effects for proper comfort. Without the intense radiant heat put off by radiators and bare tubing, occupants may be placed close to the convector and still maintain comfort.

Some units may be equipped with dampers to regulate the air flow. These are comprised of two major parts, the heating element and the enclosure. The enclosures are usually the cabinet or along-the-wall type.

Convector elements are lightweight and do not resist or absorb expansion strains that are transmitted from the riser through branch pipes. Therefore, the branch pipe should have at least three elbows.

Branch connections from risers to convectors should be arranged to maintain the proper pitch when the piping is heated and expanded. The supply connections to the heating element may be made at the bottom, top, or end of the inlet header. The return connections are made at the bottom of the opposite header. Figure E-35 shows a floor cabinet convector.

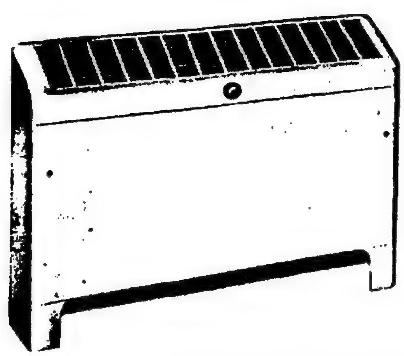


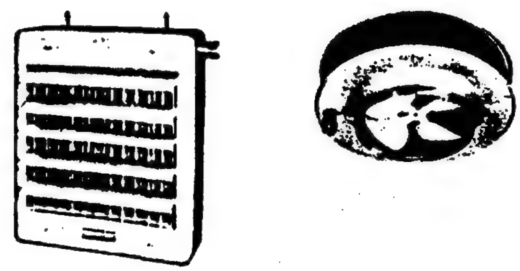
Figure E-35. Floor cabinet convector.

Illustration courtesy of Dunham-Bush, Inc.

Forced Convection Space Heating Equipment

Unit heaters are usually placed in or adjacent to the space heated to force air into the occupied space. Large-sized unit heaters are used with or without distribution ducts. The several types of unit heaters can be classified as blower fan and propeller fan types. The propeller type may be categorized as horizontal discharge and vertical discharge units.

The units are comprised of a heat exchanger and a fan enclosed in a casing arranged to be suspended from the building construction. Figure E-36 shows vertical and horizontal discharge units.



Horizontal Discharge Vertical Discharge

Figure E-36. Horizontal and Vertical Discharge Units.

Illustration courtesy of Dunham-Bush, Inc.

6 Hydronic Pumps

Centrifugal pumps are used to circulate the water in heating and cooling systems. These pumps are well adapted to hydronic systems because they are simple, compact, quiet, easy to maintain, and efficient for delivering large quantities of water against the forces encountered. Some pump applications include:

- Condenser water circuits to cooling towers and water source heat pumps
- Condensate return
- Boiler feed
- Recirculating hot water in heating systems and chilled water systems
- Circulating water to conditioning terminal units.

Pump Types

Centrifugal pumps used for heating and air conditioning may be defined by the type of impeller, number of impellers, type of casing, method of connection to driver, and mounting position. In these pumps, two types of impellers are used: single suction and double suction. A single-suction impeller has one suction or intake, while the double-suction impeller has two suctions or intakes.

Even though most centrifugal pumps used in heating and air conditioning are single suction, the significant example of a double-suction impeller is the single stage, horizontal split-case pump. Pumps with multiple impellers are called multistage pumps. Figure E-37 shows a centrifugal pump.

Like impellers, there are two types of casings for these pumps: volute and diffuser. The volute types include all pumps that collect water from the impeller and discharge it perpendicular to the pump shaft.

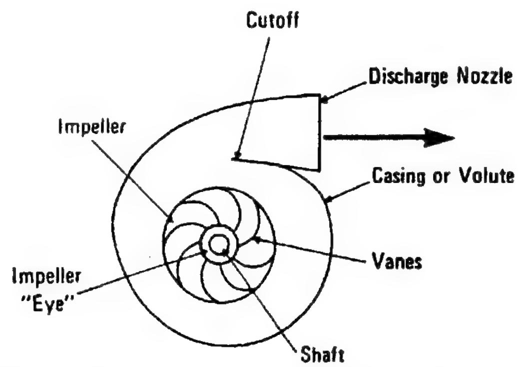


Figure E-37. Centrifugal Pump Cross-Section.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Diffuser-type casings collect water from the impeller and discharge it parallel with the pump shaft. Seven types of pumps used in hydronic systems are shown in Figure E-38. Several variations of these pumps are available from the many manufacturers who construct these pumps for particular applications.

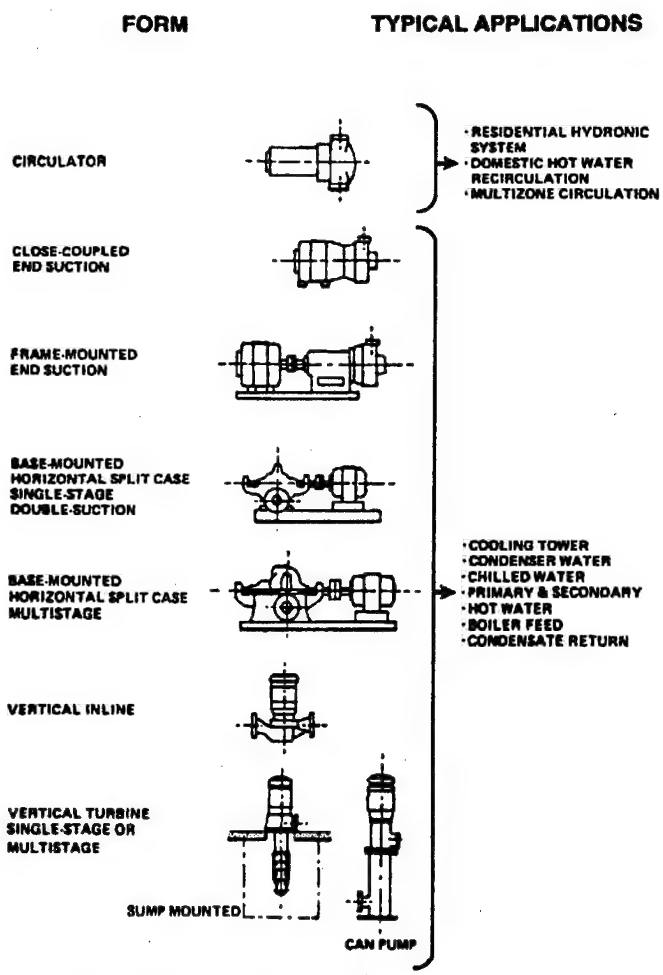


Figure E-38. Hydronic Pumps used in Hydronic Systems.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Circulators

These are low-head motor driven pumps suitable for heads in the pressure range from 8 to 30 ft at shut off. They are built to be installed in the piping and supported by it. The suction and discharge are inline and have bolted flanges for easy installation. The pipe run may be either vertical or horizontal if the shaft of the motor is horizontal. Figure E-39 depicts an inline circulator.

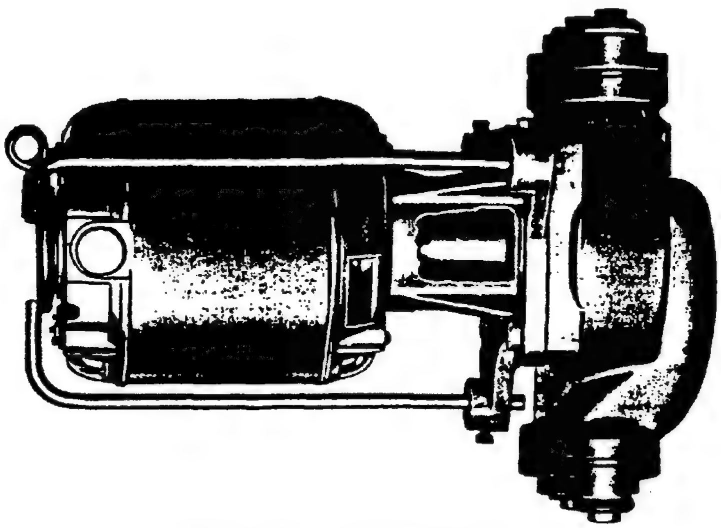


Figure E-39. Circulator (Inline).

Illustration courtesy of Dunham-Bush, Inc.

Close Coupled Centrifugal Pumps

These pumps have the impeller overhung on the motor shaft. They are compact, can operate in many positions, use simple piping arrangements, and have comparatively low cost for the capacity. An illustration of this pump is shown in Figure E-40.

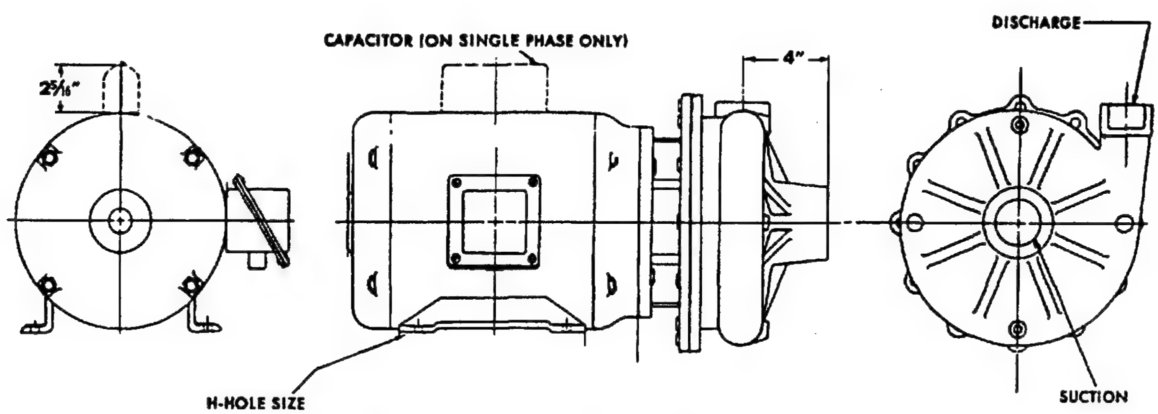


Figure E-40. Close Coupled Centrifugal Pump.

Illustration courtesy of Dunham-Bush, Inc.

Horizontal Base Mounted Pumps

The special design of this pump is for particular quietness, so hydronic systems having radiation, fan-coil units, or piping in occupied spaces may be kept at low levels of noise. The connections are flanged and the case is suitable to 125 psig. Figure E-41 shows the horizontal base mounted pump.

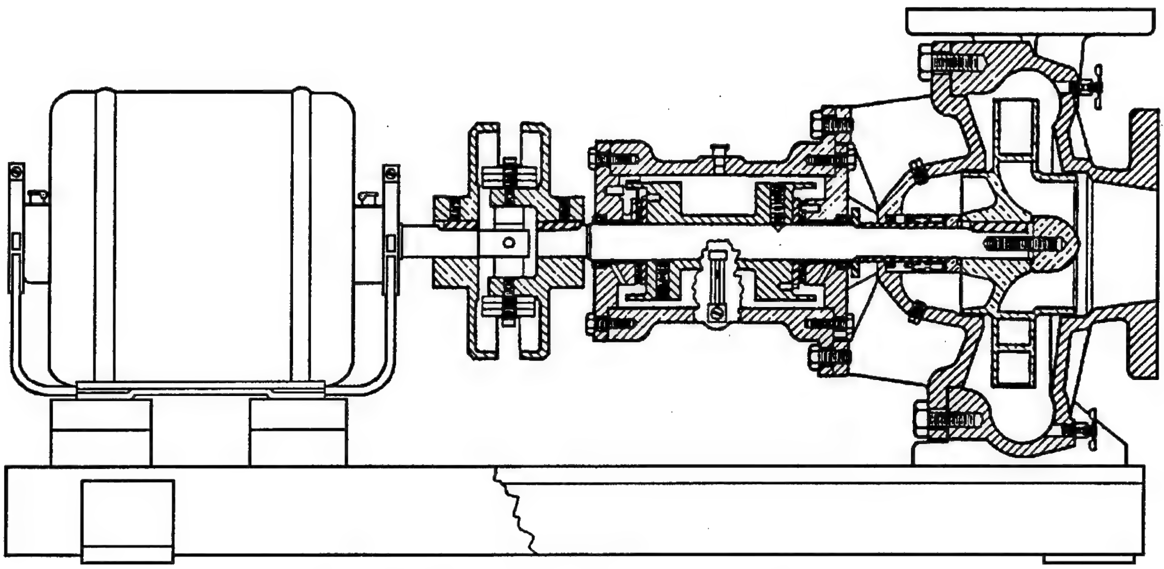


Figure E-41. Horizontal Base Mounted Pump.

Illustration courtesy of Dunham-Bush, Inc.

Horizontal Split Case Pump

These pumps (Figure E-42) usually handle larger capacities in large systems and tend to vibrate and send off low-frequency noise. With this type of pump, it is usually necessary to install an inertia base with springs. The capacity in gallons per minute for this pump is approximately 3000 gpm.

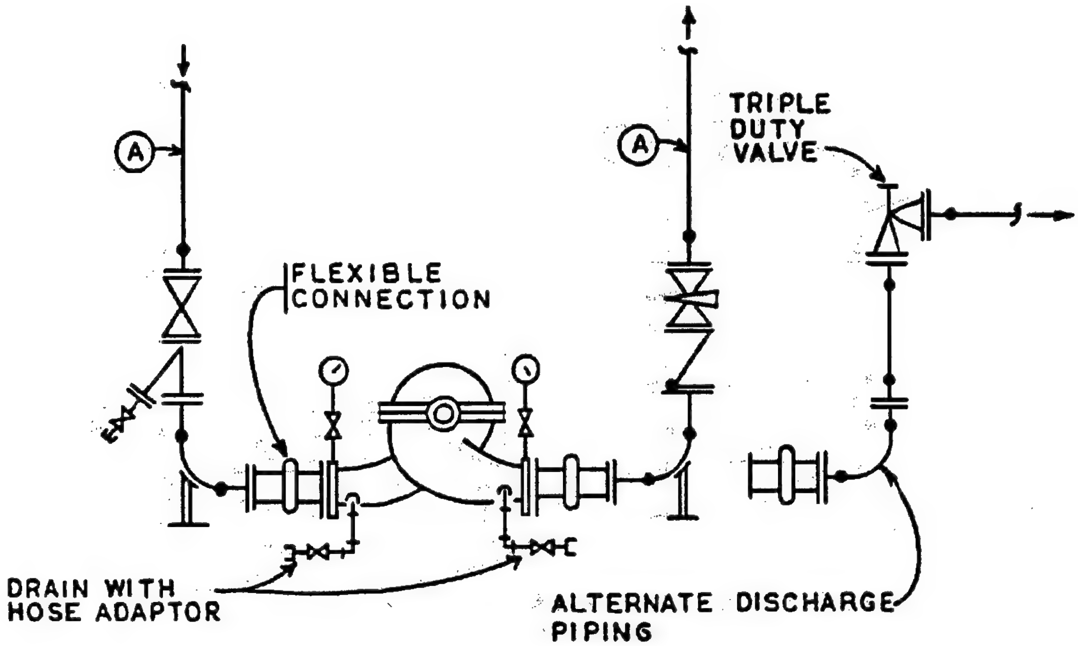


Figure E-42. Centrifugal Horizontal Split-Case Pump.

Source: Mechanical Contracting Foundation, "Guideline for Quality Piping Installation—1995," Rockville, MD. Used with permission.

Centrifugal Pump Components

Being able to identify the parts of a pump and knowing their function are the key elements in being able to identify maintenance needs. An important part of maintenance is to assure the most efficient use of all system components. The following terms will be helpful in this process:

- *Wearing rings* are for the impeller and/or casing. They are replaceable and prevent water to the impeller or casing.
- *Ball bearings* are used most frequently, except in small pumps or circulators where motor and pump bearings are the sleeve type.
- *Shaft sleeves* protect the motor or pump shaft, especially with packing.
- *Materials* that make up the centrifugal pump are generally of bronze or iron-fitted construction. In bronze-fitted construction, the impeller, shaft sleeve (if used), and wearing rings are bronze, and the casing is cast iron.
- A stuffing box is the part of the pump where the rotating shaft enters the pump casing. To seal pumps at this point, a mechanical seal or packing is used in the stuffing box.

- Balanced and unbalanced seals refer to the pressures on each side of the mechanical seal. Balanced seals are used for higher pressure seals. Inside seals operate inside the stuffing box, while outside seals have their rotating element outside the box.

- Packing is used where abrasive substances included in the water are not detrimental to system operation. Some leakage at the packing gland is needed to lubricate and cool the area between packing material and shaft.

- A balance ring is placed on the back side of a single-inlet, enclosed impeller to reduce the axial load. Double-inlet impellers are inherently balanced axially.

- Rotation is fixed by the configuration and type of vanes, and the suction and discharge connections. In addition, an arrow to indicate proper direction is often cast directly into the casing metal.

CAUTION: Pumps with mechanical seals must not be run dry, even when bumped to determine rotation.

- Operating speeds of motors usually are between 600 and 3600 rpm. The most common speed is 1800 rpm. Low speeds are generally quieter, while high speeds are less expensive.

Cavitation

Water, the fluid usually being pumped, generally contains some entrained air that has been absorbed when the fluid was exposed to the atmosphere before introduction into the system. As the fluid temperature in the system is increased, air separates out. This separation may also occur when fluid pressure is decreased. This air can be vented off somewhere in the system where there are low pressures. If fresh water is added to the system, additional venting must take place.

If air is released from the water due to low pressures and/or the liquid turns to steam, the pump (being designed for liquids) is unable to cope, and the flow of liquid is either greatly reduced or stopped completely. But at some point within the pump where the impeller produces sufficient pressure, the bubbles of gaseous liquid will be reliquified, and the bubbles of air will be reabsorbed.

The transition occurs suddenly and is accompanied by crackling or explosive noises. This noise is caused by cavitation and may cause destructive pitting and wearing of the impeller and casing as well as noise and vibration. Pump performance will be reduced by any one of these conditions.

Pump Installation Considerations

The following are some important points to remember when installing a pump:

- Suction piping should be air tight and free of air traps.
- Piping should provide a smooth flow into the suction without unnecessary elbows.
- Suction pipe should be one or two sizes larger than pump inlet.
- Reduce or eliminate restrictions at pump suction.
- Piping supported independently of pump casing.
- Use a check valve in the pump discharge piping in multi-pump installations.
- Install air vents in pump casings and piping.
- Position pressure gauges on suction and discharge at same elevation.
- Recheck pump alignment after installation.
- Lubricate prior to start up.
- Check rotation, but do not run mechanical seals while they are dry.

Pumping in Parallel

When two pumps are piped in parallel, the delivery from one pump will not necessarily be one-half the capacity of the two pumps operating together.

Each pump should be valved separately so the total capacity may be evenly divided among the pumps. Figure E-43 illustrates parallel pumps with individual valves.

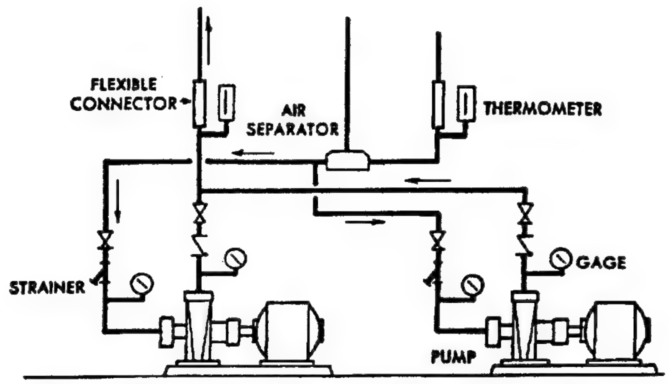


Figure E-43. Parallel Pumps with Individual Valves.
Illustration courtesy of Dunham-Bush, Inc.

When regulating the pumps, the capacity must not be so restricted that it causes an appreciable rise in temperature. This rise would increase the vapor pressure considerably. Should this happen, the pump could reach its cavitation limit and cease delivering water.

Pumping in Series

Pumps that are arranged in series discharge from one pump and deliver that same discharge into the suction end of the next pump delivering against a higher head pressure. This action is to deliver against a pressure equivalent to the sum of the heads of the two pumps. When two pumps are used for this purpose, it is called booster service.

When using pumps in series, the maximum flow will be limited by the smaller of the aggregate pumping capacity. In hydronic systems, pumps are operated in series, with one pump circulating water in a circuit supplying water to one or more other circuits or zones, each of which includes its own pump. This is called primary-secondary pumping. Figure E-44 diagrams the use of pumps in series.

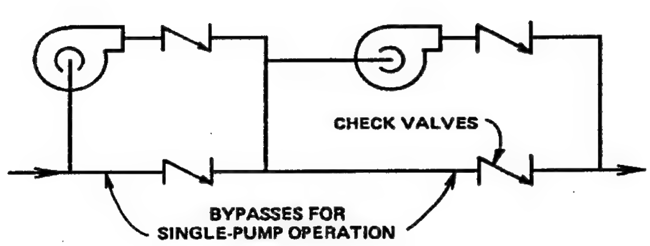


Figure E-44. Series Pumping.

7 Acceptance Testing

Before an individual or acceptance testing team begins work, it is only reasonable that an inspection of the entire system be made to confirm that all parts are on the system and ready to function. Listed below are certain key items to check before acceptance testing work begins.

Acceptance Testing Checklist

The following are items that would commonly be verified during acceptance testing. A checklist has been provided for recording data.

Coils and Heat Exchangers

- Confirm size and physical data.
- Verify air flow direction.
- Confirm provisions for pressure and temperature measurements.

Pumps

Verify:

- Equipment matches test report data (model number, make, type, etc.).
- Test report forms have had data entered that must be obtained in the field.
- All bearings have been lubricated.
- Rotation is free and correct.
- Motors have been aligned properly with pump shafts and fastened securely.

- Pump bases have been correctly grouted.
- Air has been bled from pump casing where required.
- All equipment is clean and free of foreign objects.
- Drive guards are in place.
- System pressure and temperature combinations at pump inlets are checked for possible flashing and cavitation problems.
- All strainers are clean.

The following checklist could be carried on site to perform a portion of the hydronics inspection.

HYDRONIC SYSTEM ACCEPTANCE TESTING CHECKLIST

PROJECT:______________________________

LOCATION:______________________________

NAME:______________________________

A. Coils		Correct yes	Correct no	Date Checked
1. Obstruction/debris				
2. Airflow and direction				
3. Air vents				
4. Piping connections and flow direction				
5. Pressure drop across coils				
6. Size				
7. Free of leaks				
8. Strainer installed				
9. Can coil be removed?				
10. Isolation valves installed				
11. Condition of fins				
12. Temperature across coils				
	Design	TAB	Actual	
13. Correct flow (gpm)				

B. Pumps-Motors	Correct yes	Correct no	Date Checked
1. Make, model numbers, etc.			
2. Test report forms completed			
3. Clean and free of foreign objects			
4. Rotation			
5. Lubrication			
6. Alignment/securely fastened			
7. Guards in place			
8. Pressure gauges installed			
9. Power available			
10. Disconnects installed and labeled			
11. Interlocks functional			

C. Pumps-Piping		Correct yes	Correct no	Date Checked
1. Flexible connectors				
2. Connections				
3. Pressure and temperature at pump inlet				
4. Air bled from casing where required				
5. Free of leaks				
6. Strainer clean				
7. Air vented				
8. Piping system pressure tested				
9. Pipes labeled				
10. Valves tagged				
11. Chemical treatment installed				
12. Water treatment report submitted				
13. TAB complete and approved				
	Design	TAB	Actual	
14. Correct flow (gpm)				

D. Pumps-Bases	Correct yes	Correct no	Date Checked
1. Vibration isolation			
2. Grouting			
3. Leveling			

E. Heat Exchangers	Correct yes	Correct no	Date Checked
1. Flow and connections			
2. Air vents			
3. Leakage			

F. Hydronic Piping Systems	Correct yes	Correct no	Date Checked
1. Leak tested			
2. Relief or safety valves			
3. Compression tanks and air vents			
4. Strainers clean			
5. System installed as per plans			

Basic Hydronic System TAB Procedures

The following list is a set of procedures basic to all types of hydronic distribution systems. This list originated and may be found in the National Environmental Balancing Bureau manuals for TAB. The purpose of this summary is to provide the acceptance testing team with a concise outline of what the TAB contractor was supposed to have done during TAB.

- See that all necessary electrical wiring, temperature control systems, all related hydronic piping circuits, and all related duct systems are functional, and that any necessary compensation for seasonal effects have been made.

- Determine that hydronic systems have been cleaned, flushed, refilled, and vented as required.

- Determine that manual valves are open or preset as required, and all temperature control (automatic) valves are in the normal position.

- Determine that automatically controlled devices in the piping or duct systems will not adversely affect the balancing procedures.

- With pump(s) off, observe and record system static pressure at the pump(s).

- Place systems into operation, check that all air has been vented from the piping systems, and allow flow conditions to stabilize.

- Record operating voltage and amperage, and compare these with nameplate ratings and thermal overload heater ratings.

- Record speed of each pump.

- With pump(s) running, slowly close the balancing cock in pump discharge piping, and record discharge and suction pressures at the pump gauge connections. Using shut-off head, determine and verify each actual pump operating curve and the size of each impeller. Compare this data with the submittal data curves. If the test point falls on the design curve, proceed to the next step; if not, plot a new curve parallel with other curves on the chart, from zero flow to maximum flow. Make sure the test readings were taken correctly before plotting a new curve. Preferably one gauge should be used to

read differential pressure. It is important that gauge readings should be corrected to the center line elevation of the pump.

- Open discharge balancing cock slowly to a fully open position, and record the discharge pressure, suction pressure, and total head. Using the total head, read the system water flow from the corrected pump curve established above. If the total head is higher than the design total head, the water flow will be lower than designed. If the total head is less than design, water flow will be greater; in which case, the pump discharge pressure should be increased by partially closing the balancing cock until the system water flow is approximately 110 percent of design. Record the pressures and the water flow. Check and record pump motor voltage and amperage. This data should still be within the motor nameplate ratings. Start any secondary system pumps and readjust the balancing cock in the primary circuit pump discharge piping if necessary. Again record all readings.

- If orifice plates, venturi meters, or other flow measuring or control devices have been provided in the water piping system, an initial recording of the flow distribution throughout the system should be made without making any adjustments. After studying the system, adjust the distribution branches or risers to achieve balanced circuits as outlined above. Vent air from low flow circuits. Then proceed with the balancing of terminal units on each branch.

- Before adjusting any balancing cocks at the equipment (i.e., chillers, boilers, hot water exchangers, hot water coils, chilled water coils, etc.), take a complete set of pressure drop readings through all equipment and compare with submittal data readings. Determine which are high and which are low in water flow. Vent air from low flow circuits or units and retake readings.

- Make a preliminary adjustment to the balancing cocks on all units with high water flow, setting each about 10 percent higher than the design flow rate.

- Take another complete set of pressure, voltage, and ampere readings on all pumps in the system. If the system total flow has fallen below design flow, open the balancing cock at each pump discharge to bring the flow at each pump within 105 to 110 percent of the design reading (if pump capacity permits).

- Make another adjustment to the balancing cocks on all units that have readings more than 10 percent above design flow in order to increase the flow through those units with less than design flow.

- Repeat this process until the actual fluid flow through each piece of equipment is within ±10 percent of the design flow.

- Make a final check of and record the pressures and flows of all pumps and equipment and of the voltage and amperage of pump motors.

- Where three-way automatic valves are used, set all bypass line balancing cocks to restrict the bypassed water to 90 percent of the maximum demand through coils, heat exchangers, and other terminal units.

- After all TAB work has been completed and the systems are operating within ±10 percent of design flow, mark or score all balancing cocks, gauges, and thermometers at final set points or range of operation.

- Verify the action of all water flow safety shutdown controls.

Glossary

ALCOVE: A recessed section of a room, such as a breakfast nook.

AMPERAGE: The strength of an electric current measured in amperes.

ATMOSPHERIC AIR: Air under the prevailing conditions of the atmosphere.

CONDENSATE: The liquid formed by condensation of a vapor: in steam heating, water condensed from steam; in air conditioning, water extracted as by condensation on the cooling coil of a refrigeration machine.

ENTRAINED AIR: Air that has been absorbed or suspended in water. As water heats up, air separates out and stores in high spots of a hydronic system.

GAGE GLASS: The transparent part of a water gauge assembly connected directly or through a water column to the boiler, below and above the water line to indicate the water level in a boiler. It is sometimes located in compression tanks to observe the water level.

GRAVITY CONVECTION: The transmission of heat by the circulation of a liquid or a gas such as air due to the forces of gravity.

HEAT EXCHANGER: A vessel in which heat is transferred from one medium to another.

HEAT TRANSFER: A form of energy in motion from one object to another caused by a temperature difference.

HEAT TRANSFER MEDIUM: Matter that transports energy from one body to another. The matter involved could be water, steam, vapor, or solids such as steel, copper, plastic, etc.

HYGROSCOPIC: Attracting or absorbing moisture from the air.

INTERIOR ROOM: A room with its own walls that are enveloped within another room or perimeter walls of a building.

MAKE-UP WATER: The water added to boiler feed to compensate for that lost through exhaust, blowdown, leakage, etc.

STATIC HEAD: The pressure due to the weight of the fluid above the point of measurement. The units of head are pounds per square inch or Pascals.

VENTILATION AIR: Outside air brought into the building by use of a mechanical system.

VENTING: The removal of a gas or vapor through an opening in a vessel or other enclosed space.

VOLTAGE: The electromotive force or difference in electric potential expressed in volts.

WET BULB TEMPERATURE: The lowest temperature that a water wetted body will attain when exposed to an air current. This is the temperature of adiabatic saturation.

ZONE: The specific section of a building controlled by a single thermostat. Buildings may be divided into many zones.

Bibliography

American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), *Systems and Equipment Handbook*, ASHRAE, Atlanta, GA, 1996.

ASHRAE, *Fundamentals Handbook*, ASHRAE, 1997.

Armstrong Capsulated Controlled Steam Traps, Bulletin No. 131-D10M 3/85-0, Armstrong, Three Rivers, MI.

Pressure Drop Calculations in Hydronic Systems, Bulletin No. TEH-571, File 160110, Bell & Gossett ITT, Lenexa, KS, 1971.

Heating and Cooling Coils, Burnham Corporation, Hydronics Division, Lancaster, PA.

Cash Automatic Valves ACME, Bulletin Reg. 1e, Nov. 1, 1978, A. W. Cash Valve Manufacturing Corp., Decatur, IL.

Cengel, Yunus A., Boles, Michael A., *Thermodynamics*, McGraw-Hill Inc., St. Louis, MO, 1989.

Dukelow, S.G., *Improving Boiler Efficiency*, Cooperative Extension Service, Kansas State University, Manhattan, KS, 1983.

Hydronics Manual, Dunham-Bush, Inc., West Hartford, CT, 1963.

The World Book Encyclopedia, Volume 17, Field Enterprises, Inc., Chicago, Illinois, 1951.

National Environmental Balancing Bureau (NEBB), *Environmental Systems Technology*, NEBB, Vienna, VA, 1984.

NEBB, *Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems, 4th Edition*, NEBB, 1983.

Webster's New World Dictionary, 2nd College Edition, Simon & Schuster, Inc., Springfield, MA, 1986.

CERL DISTRIBUTION

Chief of Engineers
ATTN: CEHEC-IM-LH (2)
ATTN: CEHEC-IM-LP (2)
ATTN: CECC-R
ATTN: CECW
ATTN: CEMP
ATTN: CEMP-CE
ATTN: CERD-L
ATTN: CERD-M (2)

CEISC 22310-3862
ATTN: CEISC-E
ATTN: CEISC-FT
ATTN: CEISC-ZC

US Army Engr District
ATTN: Library (40)

US Army Engr Division
ATTN: Library (11)

US Army Materiel Command (AMC)
Alexandria, VA 22333-0001
ATTN: AMCEN-F

FORSCOM
Forts Gillem & McPherson 30330
ATTN: FCEN
Installations: (20)

TRADOC
Fort Monroe 23651
ATTN: ATBO-G
Installations: (19)

Defense Tech Info Center 22060-6218
ATTN: DTIC-O (2)

109
5/99

☆U.S. GOVERNMENT PRINTING OFFICE: 1999 - 746-070/80051